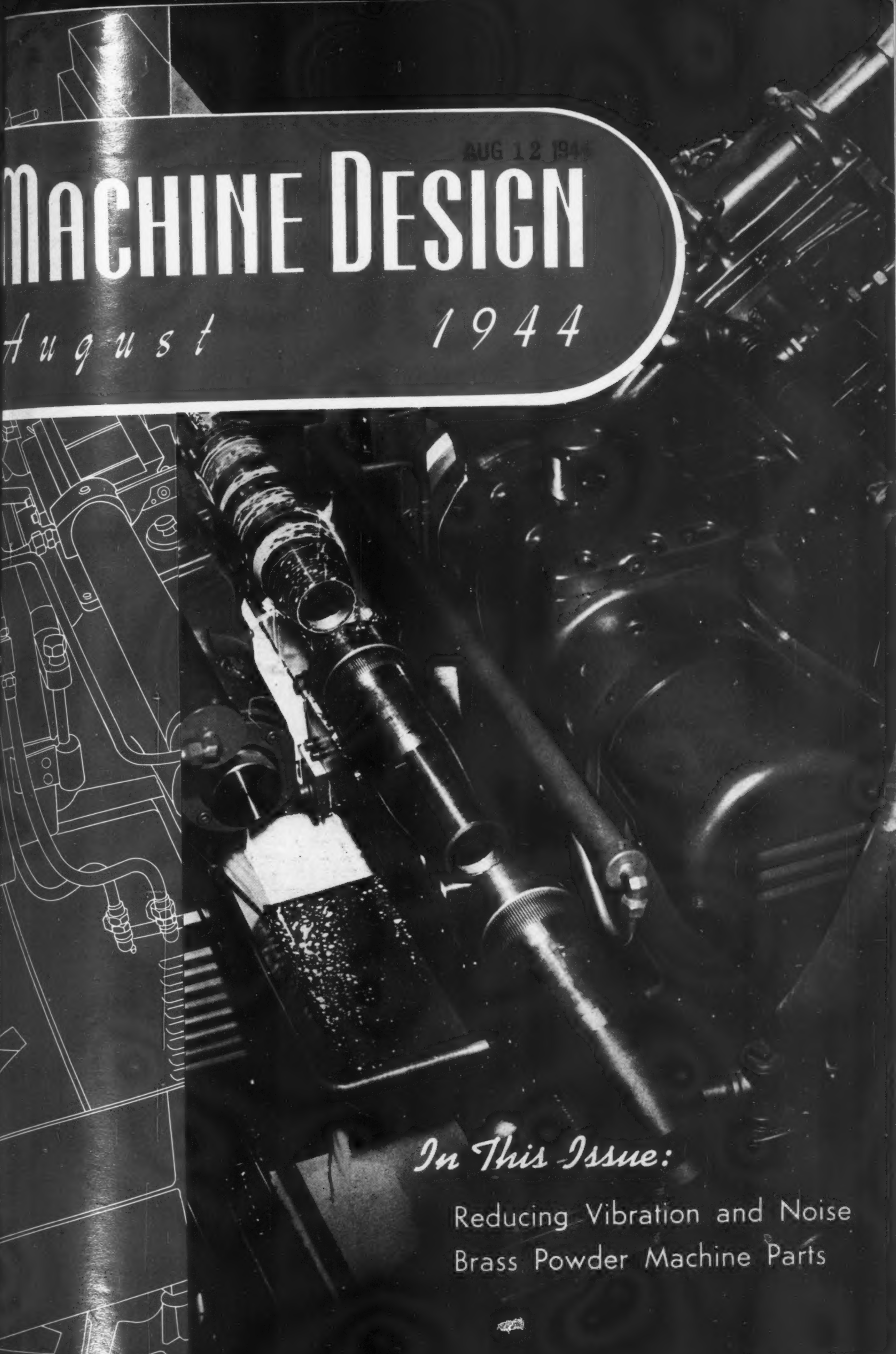
MACHINE DESIGN

August 1944



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Reducing Vibration and Noise

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MACHINE DESIGN

THE PROFESSIONAL ...

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Cover—Centerless Grinder (Courtesy Cincinnati Grinders Inc.)

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Itemized Index

Classified for Convenience when Studying Specific Design Problems

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

This issue at a glance

Excessive Vibration and Noise in Machines . . . .

. . . . often indicate faulty design. Their detrimental effects on machine performance, life and sales appeal make it imperative to hold them at a minimum. Locating sources and measuring magnitude is the subject of the timely article beginning on Page 85.

Patents May Be Good As Gold in California . . . .

. . . . yet judged invalid in New York. That's the present setup. What's needed is a more unified system in our patent courts. George V. Woodling, noted patent authority, gives his views on such a system commencing Page 93.

Postwar Designs Will Be Influenced Greatly . . . .

. . . . by recent advances in the heat treatment of steel. What are these advances? What do they offer the designer? What do they promise for the future? Harry W. McQuaid brings you vital information on this subject. Page 107.

Exceptional Strength-Weight Ratios of New Glass Laminates . . . .

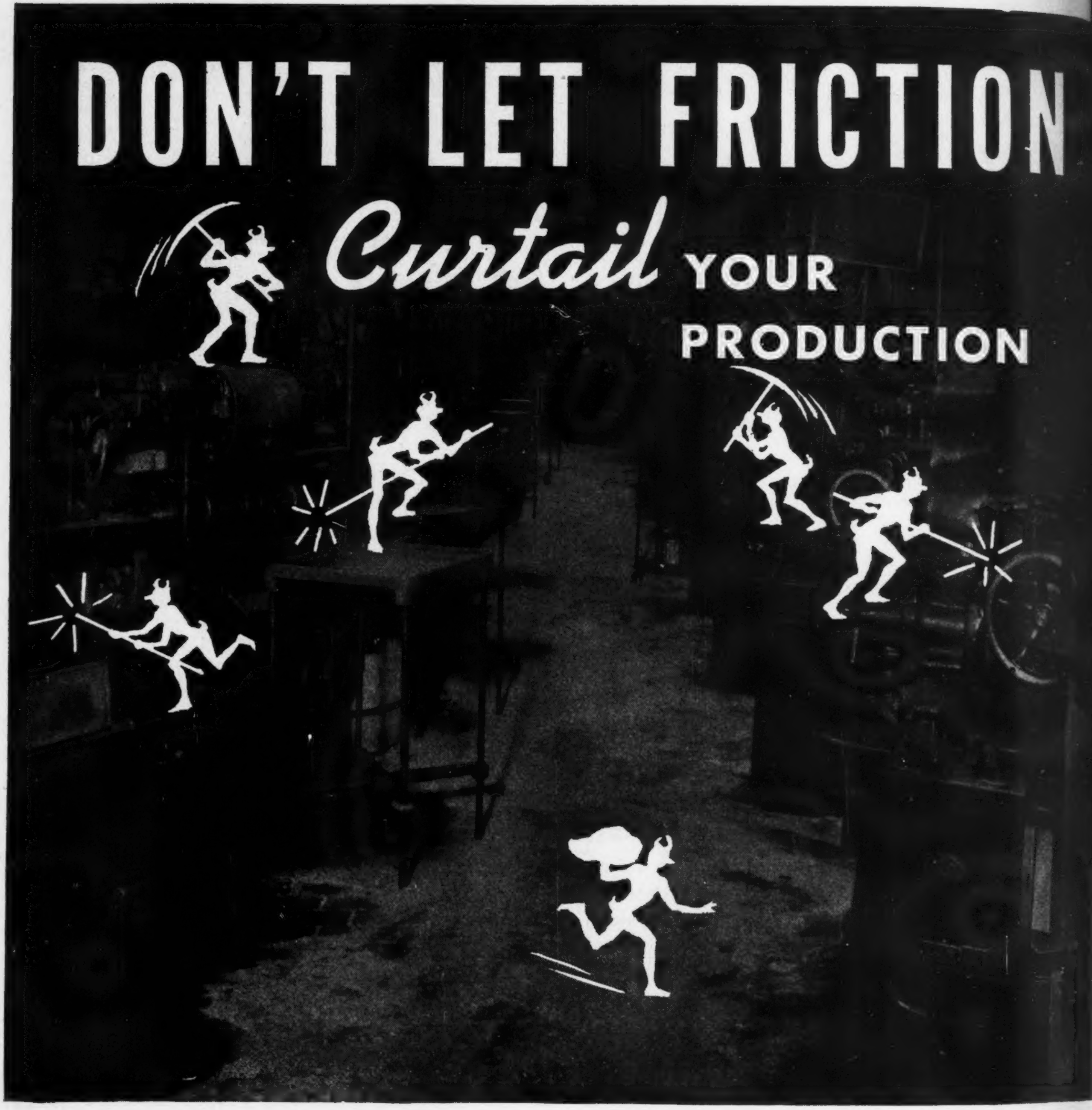
. . . . place them squarely in the running with some metals. Offering tensile strengths up to 55,000 pounds per square inch at specific gravities of 1.7 to 1.77, they're becoming increasingly important structural materials. Much of the secret lies in their bonding resins. Page 119.

Greater Use of Combination Motor-Control Units . . . .

. . . . is an ascendant trend. These compact "packages" of controlled power save space and weight—make for neater, more efficient design. Their impact on design of the future will be considerable. See Page 123.

Brass Powder Parts Can Be Designed Competently . . . .

. . . . only when their production process is understood. Shape of part not only influences feasibility of production but has much to do with strength and ductility. For the latest data on brass powder machine parts, turn to Page 129.



In the high-speed drive for Victory, FRICTION finds its greatest opportunity. Machine operators have rest intervals, but the machines themselves have none. And so, normal inspections and adjustments may be overlooked—lubrication may be neglected—a bearing becomes dry or dirty, and over-heated—FRICTION'S agents get in their deadly work—and A DAMAGED, BURNED-OUT OR RUINED BEARING CRIPPLES A VITAL TOOL.

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To Win the War:—Work—Fight—Buy War Bonds

NORMA-HOFFMANN BEARINGS CORP'N., STAMFORD, CONN. . . . FOUNDED 1911

Topics

SHRAPNELING of small gas cylinders such as used for oxygen and carbon dioxide on aircraft is prevented by winding the cylinders tightly with high-tensile steel wire so that a bullet leaves a clean hole without fragmentation. The method was developed by Walter H. Kidde and Co. Untreated cylinders are apt to explode when hit by flak or bullets because of internal pressure.

ONE COAT WHITE porcelain enamel applied direct to steel with only one firing has been developed by Pemco Corp. The process requires no special bond, pickling equipment nor special handling and the enamel can be fired at 1500 degrees Fahr., producing a brilliant opaque finish.

MODULATING LIGHT consisting of a high-pressure mercury vapor lamp with associated controls has been developed by Hanovia Chemical and Mfg. Co. to provide a steady light source for printing sound track on film or where light intensity is desired to be automatically adjusted to various levels.

GARAND RIFLE is equal to approximately 2½ times the firing power of any other nation's bolt-action rifle.

CONTINUOUS HARDENING of steel bars by high-frequency induction heating is now on a commercial basis. Bars so hardened reveal improved machinability while heat-treatment costs are reduced in most cases by 50 per cent, according to The Ohio Crankshaft Co.

SHOOTING MORE LEAD more accurately at a more distant target than any other plane ever built the B-29 gunfire system enables the plane to fly through enemy planes without fighter escort. This is made possible through new power turrets and multiple-gun installations with computing sights which automatically correct for various factors such as wind and plane velocity while putting the sight direct on the target. In a matter of seconds most of the armament can be swung about to concentrate terrific fire on one spot.

MORE THAN FOUR BILLION feet of insulated wire, an essential of front-line communications, has been manufactured by U. S. Rubber Co. In production diamond dies reduce the wire to desired size before it is run through a tinning process which protects it against corrosion from the latex outer insulation. Spark tests detect any imperfections in the product.

AIRCRAFT, engines and propellers manufactured by Curtiss-Wright during 1943 cost $1,295,236,317, an increase of 68 per cent over 1942 production. This is the first time that any aircraft firm has exceeded a billion-dollar shipment in one year, and the amount equals 15 per cent of the industry's total dollar output.

SYNTHETIC RESIN has been developed by Westinghouse to replace scarce shellac and it is believed that it will also replace mica for many applications. A one-inch bar of the synthetic would support a 40-ton load.

RUST PREVENTATIVES which are actually metal cleaning solvents provide rust-inhibited surfaces for 30 days or more. A dip method developed by Colonial Alloys Co., the films protect threaded parts, joints, interstices and crevices.

SYNTHETIC RUBBER for the thirty bullet-sealing fuel tanks of the Boeing B-29 involves nearly two tons of the material. Although the total weight of rubber in this plane is nearly 2½ tons, the tires and tubes have been redesigned to weigh less than 600 pounds and are the lightest per pound of load in any plane.

Vibration and Noise—Causes and Cures

By Colin Carmichael

Fig 1 — Vibrograph held against vibrating part of machine traces a record of the vibration on a celluloid ribbon for subsequent analysis

Part I—Measurement

ACCOMPANYING the violence of modern warfare the amount of vibration and noise which the human system must endure probably is reaching an all-time high. The seriousness of the physical effects of vibration and noise will, as never before, be brought to attention by the permanent injuries, such as deafness, which a large number of our returning soldiers will suffer. In the postwar world smoothness and quietness in machines will be demanded and in many cases may be the deciding factor in the user's mind when considering competitive equipment. New lightweight designs will be particularly susceptible to vibration and will need the most careful development to avoid objectionable characteristics. The present series of articles, of which this is the first,

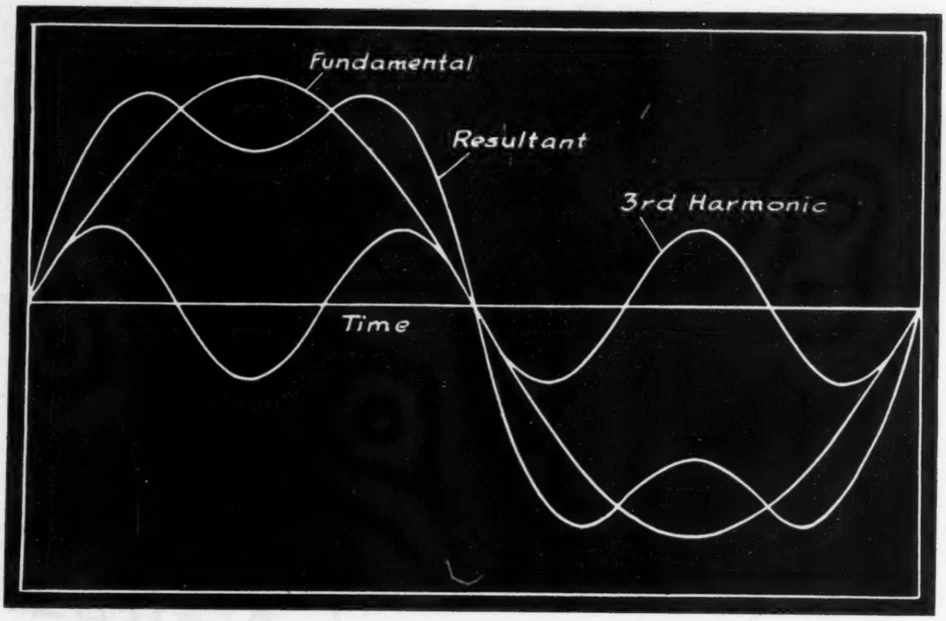


Fig. 2—Left—How a complex vibration (resultant) is composed of simple harmonic motions (fundamental and third harmonic) added together

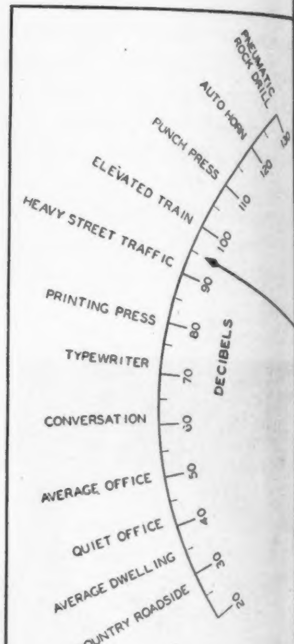


Fig. 3—Right—Sound level is measured on the decibel scale which is pictured here in terms of familiar noise sources

is a practical discussion of how vibration and noise can be tracked down and eliminated or at least reduced to satisfactorily low levels. Because of their complex nature, many types of vibration cannot be wholly anticipated in design and it often is necessary to modify the design or apply special parts or materials during the development of experimental models. In this work the assistance of suitable instruments is an invaluable aid. Inasmuch as several types of equipment are available, it is important for the design or development engineer to know what particular characteristics best meet the needs of his problems. With this in mind, the introductory article is concerned primarily with the nature of vibration and sound and with available measuring and analyzing equipment.

Nature of Vibration, Sound and Noise

Vibration means, in general, a periodic motion, although the term ordinarily is applied in a more restricted sense to the motion of solids. Vibrations transmitted to the air become sound, and noise is undesired sound. Typical of the nature of vibration is the series of curves shown in *Fig.* 2. Ordinates may denote displacement of particles, air pressure, velocity, acceleration, or any of several other measurements associated with vibration. It will be noted that the curve marked "resultant" has a complex shape which repeats itself and therefore is called periodic. Actually it is the resultant of two simple harmonic motions one of which, called the fundamental, repeats with a frequency equal to that of the resultant, while the other, called the third harmonic, repeats with a frequency three times that of the fundamental. Some instruments indicate or record the exact shape of the resultant curve, some indicate the effective height of the resultant while others indicate the effective heights and frequencies of the separate components.

Actual vibration or sound waves are much more complex than the simple example illustrated, and may contain a large number of harmonics or "overtones". Sound waves which are of this type are called "pitched sound" because definite pitch (the term used to denote the effect of sound frequency on the ear) can be recognized. Hum

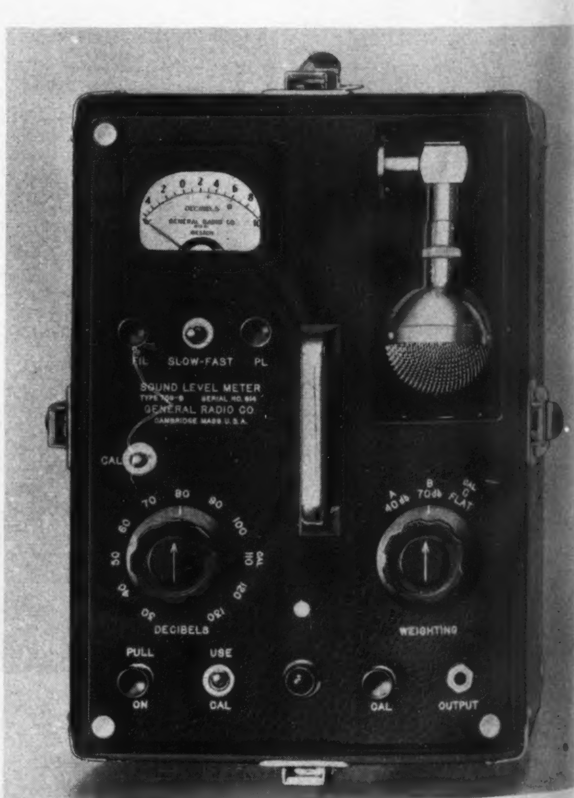


Fig. 4—Sound-level meter may be used with microphone attached to cabinet as shown, or remote from microphone

r whine in a machine is an example of pitched sound. The distinguishing characteristic is that the sound waves on analysis are found to contain only certain definite frequencies which bear simple ratios to the fundamental. In "unpitched sound" such as clatter and din no definite musical tone is recognizable because many frequencies unrelated to each other are present.

How the Ear Responds to Various Frequencies

Sound frequency, as previously noted, is perceptible to the ear as a tone. On a piano the frequency of the lowest tone is 27.5 cycles per second while that of the highest is 4186 cycles per second, this representing approximately the range in which the ear is most sensitive. Sounds of considerably higher frequency are detected by the average ear usually as harmonics or overtones rather than as fundamental tones. These overtones are perceived by the listener through the quality of the sound and tell him whether the tone is emitted by a violin, an air-raid siren, a high-speed machine, or some other recognizable sound source.

In addition to perceiving sound frequency, the ear is sensitive to sound intensity. It is not, however, capable of assigning as accurate a value to sound intensity as it can to frequency; hence the need for an instrument capable of furnishing objective measurements. Sound intensity is proportional to the power in a sound wave at any point, and usually is measured in watts per square centimeter. Because of the approximately logarithmic nature of the ear's response to sounds of different intensities, practical measurements of sound intensity are reported in terms of sound-intensity level, in decibels. By definition, sound level in decibels is ten times the logarithm (to the base 10) of the ratio of the sound intensity to a reference intensity. The American Standard reference intensity is 10^{-16} watts per square centimeter, which is less than a billionth of a billionth of a horsepower per square inch, and is perceptible only to

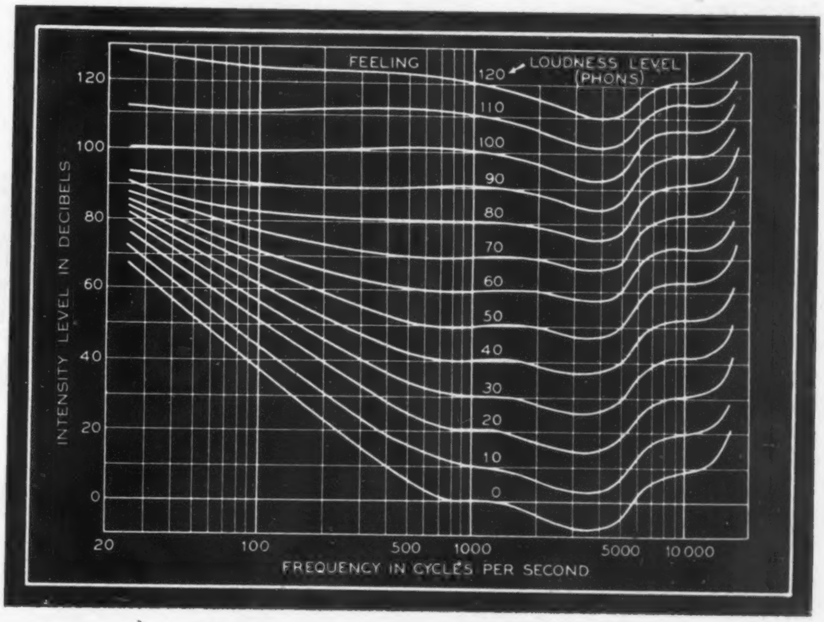
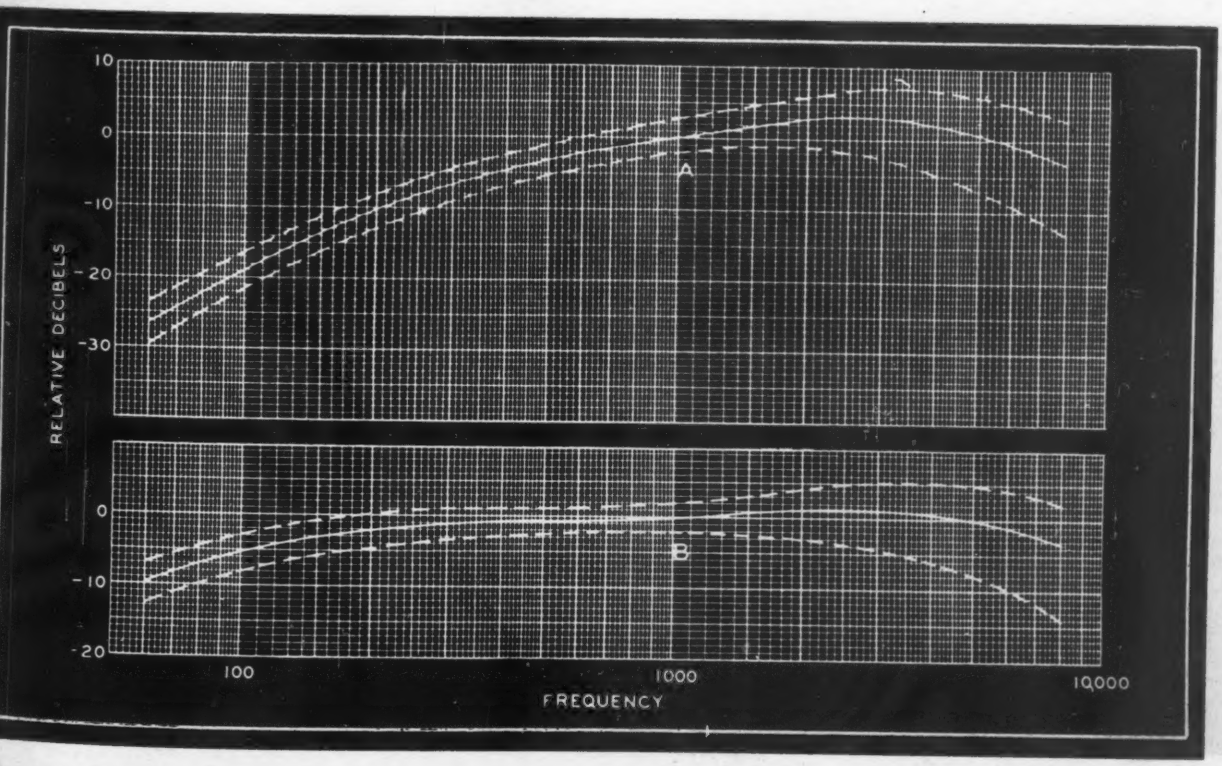


Fig. 5—Right—Curves of equal loudness level as perceived by the average ear are incorporated in A.S.A. standards

Fig. 6 — Below — Frequency-response curves used in sound-level meters approximate the ear's response at 40 decibels sound level (curve A) and 70 decibels sound level (curve B). Curves, including tolerances denoted by dotted lines, are standardized by the A.S.A.



the most acute hearing. The decibel scale, in terms of familiar sounds, is graphically illustrated in *Fig.* 3.

Inasmuch as sound is the effect of pressure variations in the air, the decibel scale may also be expressed in terms of pressure. Sound intensity is proportional to the square of pressure, hence the sound-pressure level, in decibels, is twenty times the log (to the base 10) of the ratio of effective pressure to the reference pressure. Thus,

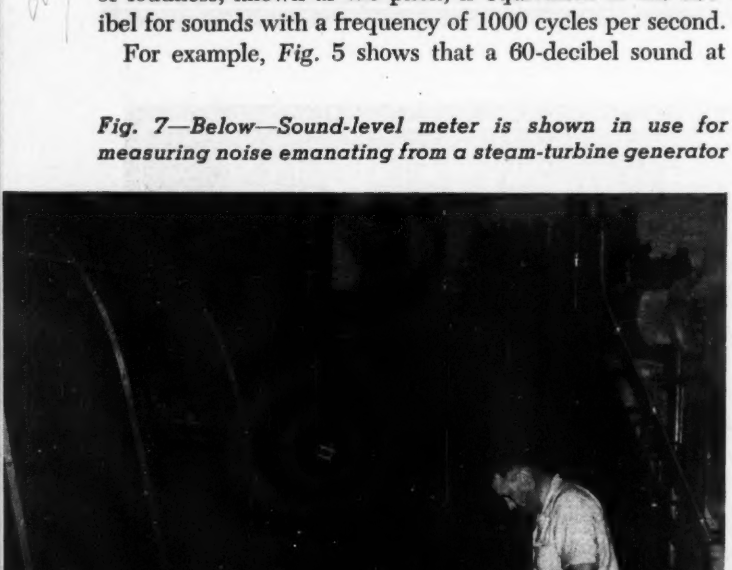
$$Decibels = 10 \log_{10}\frac{I}{I_0} = 20 \log_{10}\frac{P}{P_0} \quad \text{(1)}$$

where I denotes intensity and P denotes pressure. The reference pressure, P_0, corresponding to zero on the decibel scale, is .0002 dynes per square centimeter, which is 2.9 billionths of a pound per square inch. If it were possible to produce sound waves of such violence that the fluctuations went from zero (a perfect vacuum) to twice atmospheric, the corresponding intensity level would be approximately 190 decibels, which therefore represents the upper limit of the scale. The intensity then would be nearly nine horsepower per square inch.

Sound level may be measured with the aid of a sound-level meter, *Fig.* 4, which includes a microphone to pick up the sound waves, an amplifier to magnify the output of the microphone, and an indicator showing the decibel reading at the point where the microphone is located. Such an instrument may be made to respond equally to sounds of all intensity levels within the audible range, but the resultant scale reading would not as a rule represent accurately the noise level as judged by the average person. This is because the ear's response to sounds of various frequency is not uniform, but varies as shown in *Fig.* 5. Each curve on this chart represents constant loudness as judged by the average observer. The unit of loudness, known as the phon, is equivalent to the decibel for sounds with a frequency of 1000 cycles per second.

For example, *Fig.* 5 shows that a 60-decibel sound at

Fig. 7—Below—Sound-level meter is shown in use for measuring noise emanating from a steam-turbine generator



100 cycles per second seems no louder than a 40-decibel sound at 1000 cycles per second. Because of this characteristic of the human ear, sound-level meters are provided with a choice of weighting networks which can be employed to approximate the ear's response at various sound levels. For very high intensity levels, it is apparent from *Fig.* 4 that the response of the ear is comparatively uniform; hence for sounds in this range a network giving a flat response is used. For other levels satisfactory results are obtained by using either of two other networks, which approximate the ear's response at 40 decibels and 70 decibels respectively. Commercial sound-level meters are provided with these networks, in accordance with the American Standard curves shown in *Fig.* 6 as well as the flat response.

When making noise observations on machines which are too large to be placed in a sound-proof room, as in *Fig.* 7, the question of background noise crops up. Of the decibel reading on the scale, how much is due to the machine and how much to noises that would be present anyway? Two readings at the same location, one with the machine running and one with it stopped, will of course supply the answer, but because of the logarithmic nature of the decibel scale the subtraction cannot be performed directly. The calculations may be understood from Equation 1, which may be rewritten

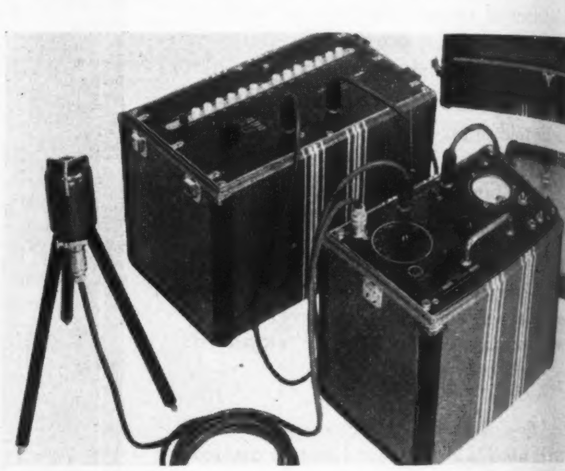


Fig. 8—Simplified noise analyzers of this type are employed for determining the magnitude and general frequency composition of a wide variety of noises

$$I = I_0 \, antilog \frac{d}{10} \quad \text{(2)}$$

where d = the sound level in decibels. Since the intensities, I, can be subtracted or added directly, it is possible to write

$$I_a = I_c - I_b \quad \text{(3)}$$

where I_a = intensity of noise to be measured, I_c = combined intensities due to machine and background, and I_b = intensity of background noise. Substituting values from Equation 2,

$$antilog \frac{d_a}{10} = antilog \frac{d_c}{10} - antilog \frac{d_b}{10} \quad \text{(4)}$$

which may be used to determine the required value d_a. In practice the correction for background noise is unnecessary if the two readings d_c and d_b differ by more than 10 decibels.

A modification of Equation 4 may also be used to determine the resultant sound level due to two or more sources of equal or unequal intensity. In applying the decibel scale it is important to keep in mind that a sound which is 10 decibels higher than another actually has ten times the intensity, while an increase of 3 decibels represents double the intensity or sound power. If the noise from a single machine, for example, measures 75 decibels sound level at a certain point, that from two such machines at the same distance would be 78 decibels and that from ten would measure 85 decibels. Again, as the distance from a noise source changes, the sound intensity varies inversely as the square of the distance; hence if the distance is doubled the decibel reading decreases about 6 decibels. These and other factors such as the influence of walls and surroundings in reflecting sound are important considerations in using sound-level meters.

While the sound-level meter is invaluable in determining whether a satisfactory degree of quietness has been attained, and is much more sensitive to slight changes than is the ear, the information which it furnishes must in many cases be supplemented by that from an analyzer, *Fig.* 8, if the source of the disturbance is to be found.

The noise from a complex machine or mechanism may come from many sources, such as loose parts, impacts, gas or liquid pressure fluctuation, etc. Location of the most objectionable disturbances often is a highly elusive problem. A sound analyzer furnishes exact information concerning the frequencies and sound levels of all important noises emanating from a machine. Whereas the sound-level meter indicates merely the effective pressure of the resultant wave, *Fig.* 2, the sound analyzer tells the frequency and amplitude of each of the component waves, such as the fundamental and third harmonic illustrated. Inasmuch as the frequencies of certain motions in the machine are functions of its speed, they are known and—if guilty—reveal themselves in the analyzer just as surely as a fingerprint incriminates a thief or murderer.

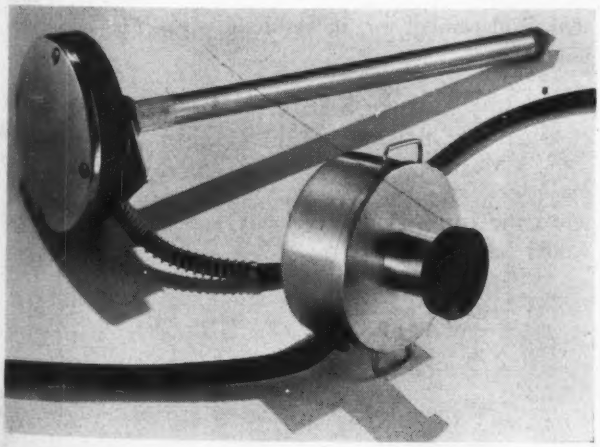


Fig. 9—Vibration pickups include the displacement type, foreground, and the inertia or acceleration type, back

In specifying and applying noise analyzers an important consideration is the width of the frequency band, or sharpness of tuning. It will readily be understood that razor sharpness in tuning to a particular frequency is neither possible nor desirable, and in practice the analyzer at any setting indicates the total intensity level of all sounds having frequencies within a certain band. For example, if the width of band at 100 cycles per second is 10 cycles, all frequencies between 95 and 105 will be included.

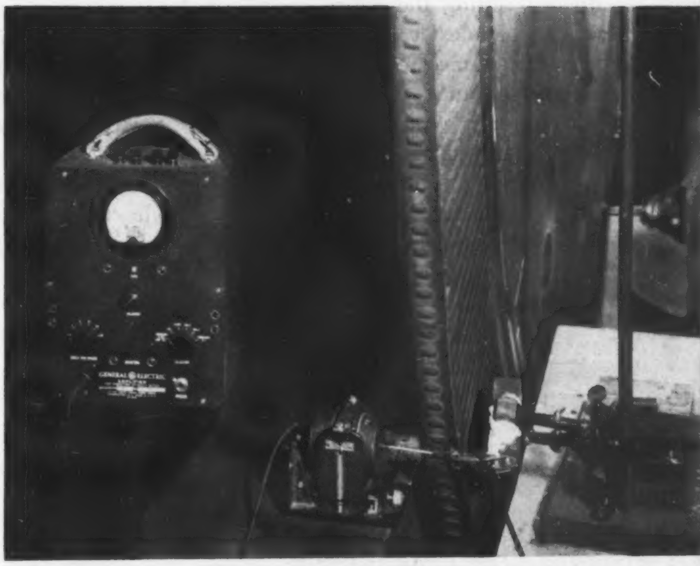


Fig. 10—Vibration-velocity pickup, shown touching turbine blade, combines with vibration meter to reveal vibration characteristics of assembly under artificial vibration

When measuring machine noises which do not stay exactly on pitch and which contain a fair percentage of unpitched sound, it will be recognized that a relatively wide band is highly desirable. Noise analyzers with wide band-pass characteristics for the purpose described are so designed that the band width is a constant percentage of the frequency being analyzed. In analyzers of this type, an example of which is shown in *Fig.* 8, selection of any one of a number of bands is made by operation of a multiple pushbutton switch, the ratio of frequencies at the upper and lower ends of each band being approximately two to one. For each band the sound level is indicated in decibels. In addition to serving as an analyzer this type of instrument also can be used as a sound-level meter, in which case the band width is, in effect, the whole range of audible frequencies. Appropriate weighting networks (*Fig.* 6) are used, depending upon the sound level.

When Sharp Tuning Is Desired

For extremely accurate analysis (narrow band width), as is desired in certain types of machines where noise output consists primarily of pitched sound of unvarying frequency, heterodyne instruments are available. In these the band width is constant at all frequencies; hence the tuning at high frequencies is extremely sharp. For example, a band width of 5 cycles, which is standard in some instruments, is only one-twentieth of one per cent of the upper frequency of 10,000 cycles per second. This type of analyzer also can be supplied with built-in filters giving a selection of wider bands such as 20, 50 or 200 cycles. Unlike the wide band pass analyzer, the narrow band analyzer has virtually stepless change, hence is used somewhat differently. By continuously varying the fre-

quency setting the entire range can be scanned rapidly and the loudest components noted, after which accurate tuning and measurement of each can be made. If the complete sound spectrum is desired, a recording unit which traces a record of the decibels at each frequency may be employed.

Vibration measurement and analysis follow the same general principles as in the case of sound. Chief differences lie in the lower frequencies involved, in the substitution of a vibration pickup for the microphone, and in the need for obtaining results in terms of displacement,

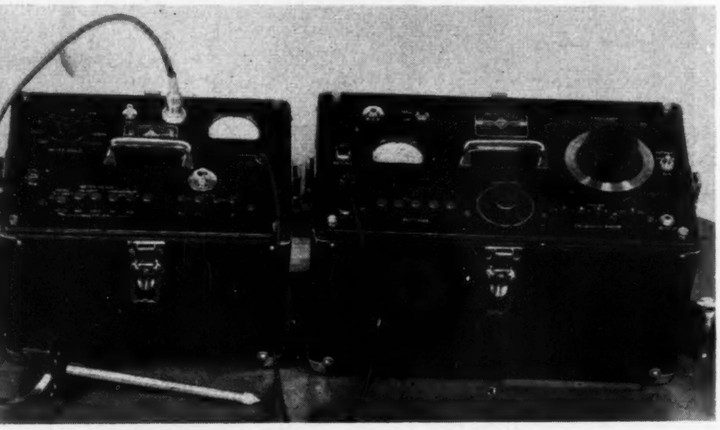


Fig. 11—Above—Vibration meter, left, working in conjunction with analyzer shown at right sorts out the various harmonics in a complex vibration

Fig. 12—Below—Amplifier unit, left, and direct-inking oscillograph, right, when used with the appropriate pickup furnish a record of the shape of a vibration wave

velocity or acceleration instead of decibels. This last point is an important one, and to explain it requires a brief excursion into the realm of mathematics. As an example, the resultant curve in *Fig.* 2 may be assumed to represent the displacement of a vibration, that is to say, the actual movement of the vibrating part. Then at any time *t* the displacement is the sum of the displacements of the fundamental and third harmonic, which may be written

$$x = A_1 \sin \omega t + A_3 \sin 3\omega t \qquad (5)$$

where A_1 and A_3 are, respectively, the amplitudes of the fundamental and third harmonics, and ω is the frequency in radians per second, which is equal to $2\pi f$, f being the frequency in cycles per second.

Velocity of the vibrating motion is found by differentiating with respect to time:

$$v = \omega A_1 \cos \omega t + 3\omega A_3 \cos 3\omega t \qquad (6)$$

and acceleration results from a second differentiation:

$$a = -\omega^2 A_1 \sin \omega t - 9\omega^2 A_3 \sin 3\omega t \qquad (7)$$

Comparison of Equations 5, 6 and 7 shows that the wave shapes of the x, v and a curves are different, the velocity curve having a third harmonic three times that of the displacement curve, and the acceleration one of nine times. Higher harmonics, of course, are even more exaggerated in the velocity and acceleration curves. Vibration measurement and analysis equipment may be made to respond to displacement, velocity or acceleration. Which of these is used naturally will vitally affect the readings either of the vibration meter or of the analyzer.

A vibration meter suitable for all but the extremely low frequencies must be capable of furnishing a steady reading of a fluctuating quantity. Inasmuch as the vibration varies periodically from plus to minus its average is zero, hence the measuring equipment must include means for converting to all positive values. This is done by squaring the ordinates of the periodic curve in *Fig.* 2. By taking the square root of the average reading so obtained, the effective value called the root-mean-square (rms), is determined. For a simple harmonic wave this value is equal to the amplitude divided by the square root of two, for example, $A_1/\sqrt{2}$. Vibration-meter readings of effective displacement for the resultant wave, *Fig.* 2, therefore would be

$$x_{rms} = \frac{1}{\sqrt{2}}\sqrt{A_1^2 + A_3^2} \qquad (8)$$

while the corresponding velocity and acceleration values would be

$$v_{rms} = \frac{\omega}{\sqrt{2}}\sqrt{A_1^2 + 9A_3^2} \qquad (9)$$

$$a_{rms} = \frac{\omega^2}{\sqrt{2}}\sqrt{A_1^2 + 81A_3^2} \qquad (10)$$

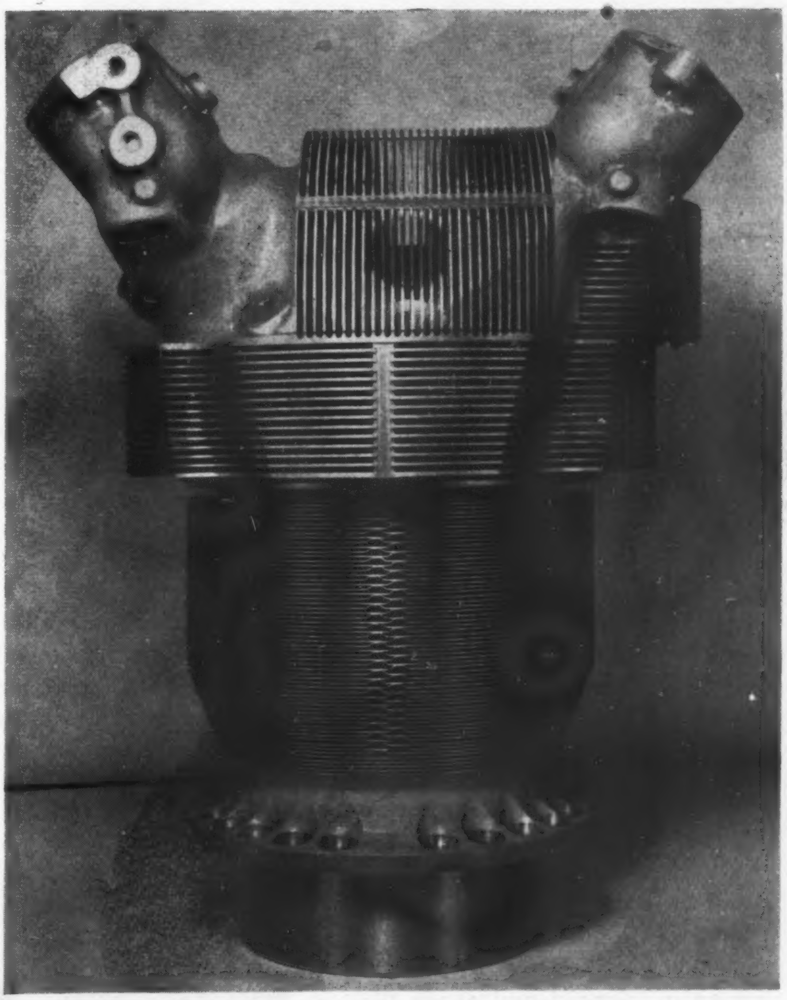
Comparison of these three values indicates the degree of difference to be expected with the three types of measurement.

Vibration pickups are designed to respond to displacement, to velocity or to acceleration. Those illustrated in *Fig.* 9 respond to displacement and acceleration while the one in *Fig.* 10 responds to velocity. Vibration meters and analyzers are provided with integrating circuits so that displacement and velocity measurements may be made with an acceleration pickup, for example. Differentiating circuits permit the use of a velocity pickup to

(*Concluded on Page* 176)

Scanning the Field for Ideas

ALUMINUM cooling fins on steel aircraft cylinders as shown at right provide almost twice the cooling area formerly obtained from machined fins cut from the barrel itself. Permitting the use of greater horsepower output as well as saving one pound of weight per cylinder, the fins are produced by folding sheet aluminum into an exaggerated W-shape with the outside legs one inch long and the center portion one-eighth-inch high. Strips are cut to size, shaped into semicircles and snapped into shallow dovetail grooves cut into the barrel. A special tool then rolls over the short center legs into the grooves, wedging the outside legs against the groove walls. This holds the fins firmly in place and provides a broad contact area through which heat can pass from the cylinder to the fins. These aluminum fins designed by Wright Aeronautical in cooperation with Scandia Manufacturing Co., can be made to almost any desired height but the steel fins had virtually reached their maximum depth (⅝-inch) to which they could be cut in quantity production.



Torque pulsations of internal combustion engines are simulated by Pesco Products Co. in testing hydraulic pumps in their laboratory as shown in the photograph at left. In this setup a universal joint is adjusted angularly to produce torsional pulsations such as found on an aircraft-engine drive. Solenoid valves and electric timers permit operation at high or low pressures in conformance with test specifications. Speed variations are obtained through variable cone pulleys connected by a wood-block belt, giving a large number of speeds at a rapid rate with fine adjustment.

Masking devices for plating or painting as shown at left are molded rubber or synthetic. They may be reused repeatedly thus conserving time in masking and stripping as well as obviating the tedious task of cleaning which is associated with other methods. Developed by Duggan Masking Devices these masks, designed to be liquid tight, can be cleaned readily in a solvent. They may also be used in dipping or spraying operations.

Self - aligning bushings which correct for end play, offset or misalignment through a specially designed linkage system, transfer push-pull cable operation into angular motion. This linkage design, shown at right, is capable of operation through an arc of 180 degrees. Designed by Simmonds Aerocessories Inc. the unit is completely enclosed.

Electronic sorting device, below right, separates oversize and undersize assemblies from those that are within proper tolerances. These assemblies are fed onto a slide as shown. About halfway down the slide the assembly comes to a contact point located at a preset height. If the part is oversize it touches the point and closes the grid circuit of an electronic tube, energizing an electromagnetic relay. This actuates a solenoid and sends the part down a chute into a container for oversize parts. A short distance beyond the first contact point a second contact is set at standard height less tolerance. Parts touching this point are delivered into another container as acceptable. Those which are undersize pass undisturbed into a third tray. Designed by General Electric, this sorting unit replaces tedious hand inspection with a needle micrometer.

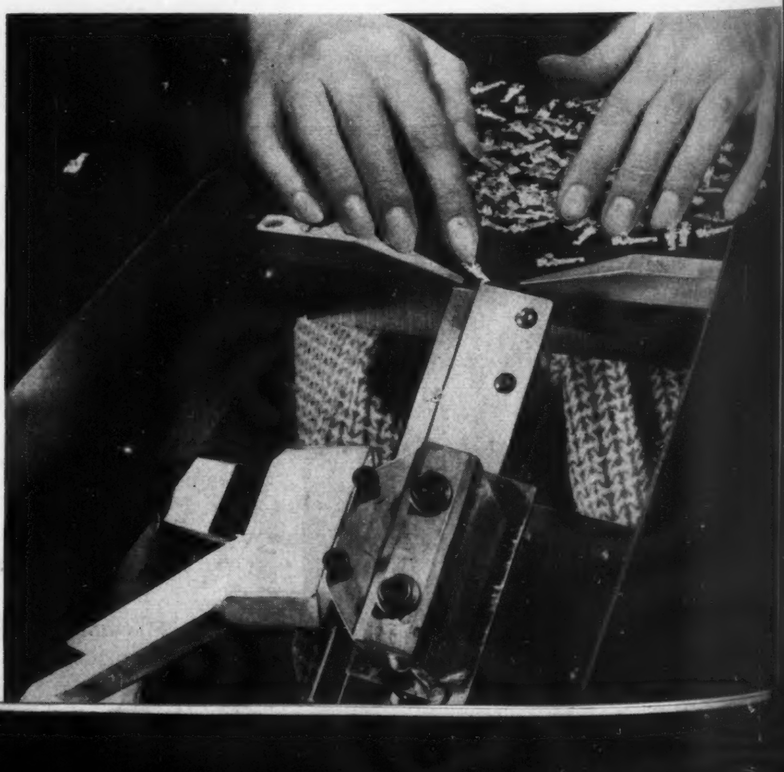


Fig. 1—Boundaries of the ten independent judicial circuits into which the United States is divided are shown on this map

Why the Patent Courts Should Be Synchronized

By George V. Woodling

SYNCHRONISM in a machine keeps all the working parts operating in harmony. Without perfect unison, trouble would occur in the overall operation and to effect uniform action the parts are geared or otherwise tied together by a common linkage.

Synchronizing the federal courts which try our patent cases is no less important that synchronizing a machine. Trouble occurs when courts render conflicting opinions, with resulting confusion which reduces the overall efficiency of the patent system. To establish uniform decisions, all the courts must be geared together by a linkage based upon a standard measurement for patentable invention. Thus, for example, the decision of a patent suit in California should be the same as that in a suit upon the same patent in New York.

In this article it is explained how the United States is divided into independent judicial regions or circuits. Further,

Fig. 2—Left—How the sixth federal circuit is subdivided into districts each having a court with one or more judges

it is shown how the synchronizing of these federal courts would more effectively strengthen our patent system than would the establishment of the proposed single court of patent appeals.

Our country, excluding the District of Columbia, is divided into ten federal circuits, *Fig.* 1. Each circuit has a separate court of appeals of three or more judges, three sitting in each case. As an example, for the sixth circuit, embracing the states of Ohio, Michigan, Tennessee, and Kentucky, the court of appeals is located at Cincinnati. Each circuit in turn is subdivided into districts, as in *Fig.* 2, which shows the subdivision of districts for the sixth circuit. Each district has a court with one or more judges. The district court located in Cleveland, which takes in the northern part of Ohio, has three district federal judges who sit singly. A party losing his case in the district court at Cleveland may appeal his case to the court of appeals for the sixth circuit at Cincinnati. The party losing in the circuit court of appeals may, in some cases, carry his case to the Supreme Court of the United States, as explained later.

A patent infringement suit is first instituted in a district court. The infringer may be sued either in the federal district of residence (if an individual) or of incorporation (if a corporation) or in the federal district in which the infringer has a regular and established place of business

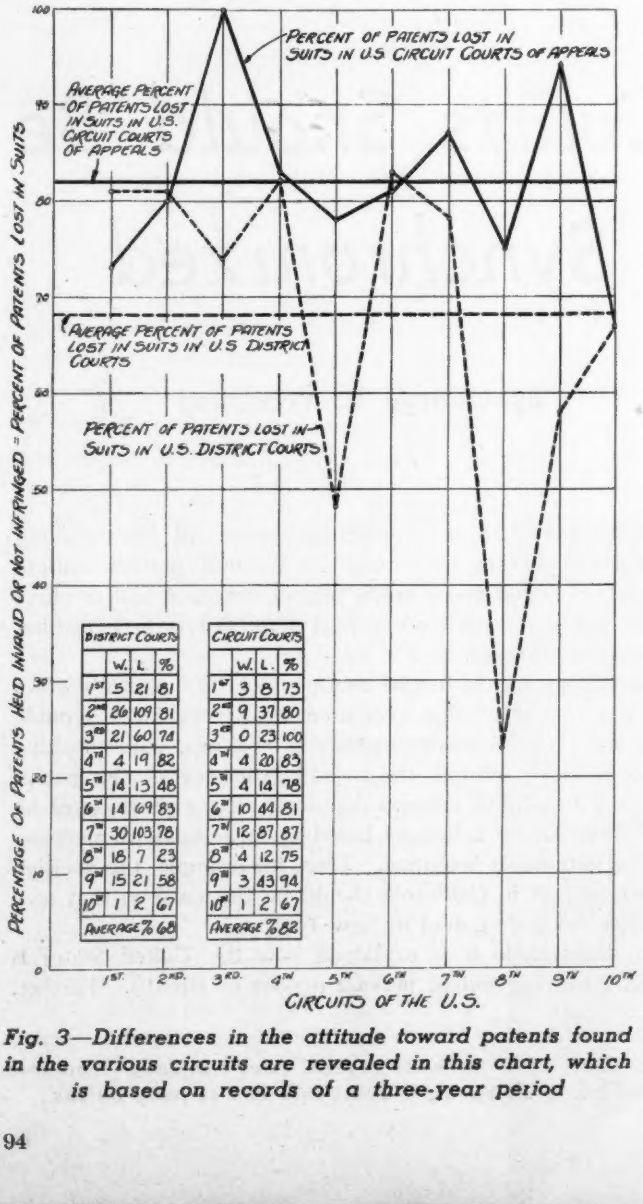


Fig. 3—Differences in the attitude toward patents found in the various circuits are revealed in this chart, which is based on records of a three-year period

and in which an act of infringement has occurred. For example, a corporation may be incorporated under the laws of New York and may be sued in the federal court located in Cleveland if the corporation has a regular and established place of business and is committing an act of infringement in the northern part of the state of Ohio.

Each Circuit Is Like a Separate Country

The ten circuit courts of appeals have final jurisdiction of patent cases and each circuit may act independently of each other upon the same patent. The several circuits are like separate judicial states or countries and the court of appeals for each circuit is supreme in its own circuit. Because of the independent operation of each circuit, the owner of a patent may bring a first suit against the X-corporation in one circuit and a second suit on the same patent against the Y-corporation in another circuit. In the first suit the court of appeals for that circuit may hold the patent valid and infringed, while in the second suit the court of appeals for that circuit may hold the opposite.

Under the present system, where there are two or more conflicting decisions on the same patent from different circuits, the case may be taken to the Supreme Court for a final determination. In taking the case to the Supreme Court, the aggrieved party petitions the court for a writ of certiorari (review) and the court in its discretion may require the circuit court of appeals to send the case up for review. Such review is not a matter of right but of sound judicial discretion, and will be granted only where there are special and important reasons therefor. One of the controlling reasons is to settle conflicts between the several circuits. Where patent infringement exists in only a single circuit, and the patent owner does not have anyone else to sue in another circuit, a conflict between two or more circuits cannot arise with respect to the patent in question. In these cases the circuit court of appeals in which the suit is brought is the final court, unless the patent is so important as to involve a great public interest, when the Supreme Court may review the case even though there is no conflict between two or more circuit courts of appeals.

U. S. Supreme Court Seldom Handles Patents

Under this system of operation, the number of patent cases tried by the Supreme Court is extremely low. Actual figures show that the Supreme Court, on the average, handles less than three patent cases a year. This means that the bulk of such cases are disposed of in the ten circuits, and in effect the ten circuit courts of appeals become ten separate supreme courts for patents. With ten such unsynchronized courts the lack of uniformity is bound to result. In patent cases, this is more critical than in other cases which involve only general law matters, because in patent cases the judges must deal both with law and engineering. It is common knowledge that most judges have considerable difficulty with engineering technical facts.

One of the strongest measures which has been discussed and recommended for legislative action to improve the administration of uniform justice for patent cases is the establishment of a single court of patent appeals. It has been suggested that this court would have exclusive ap-

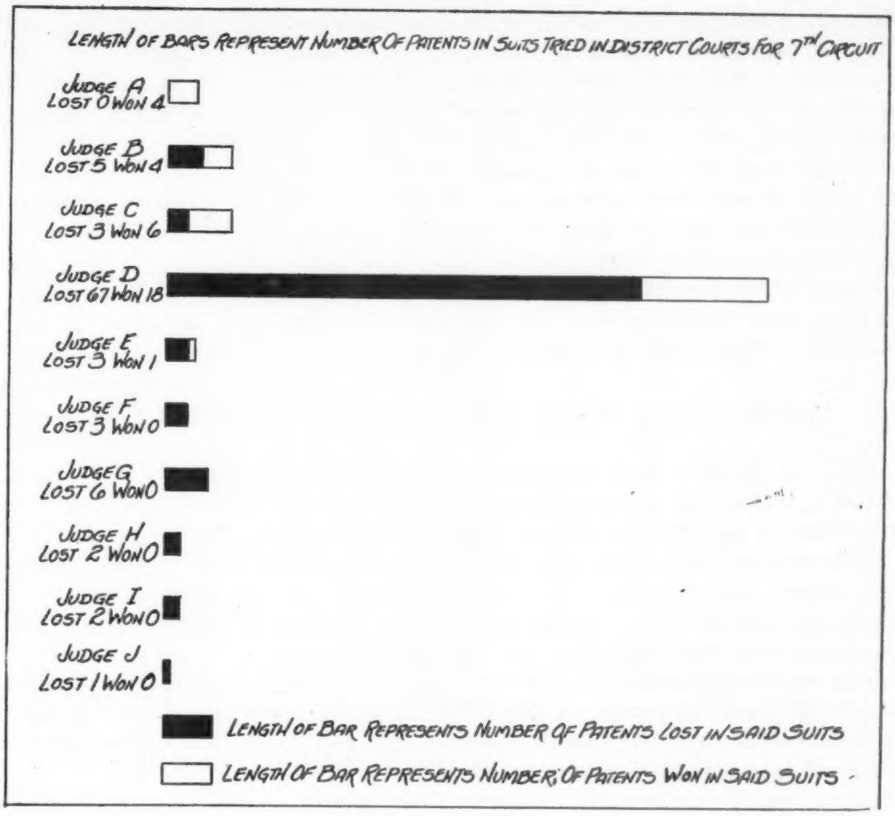


Fig. 4—Breakdown shows that patentees lost their cases in 78 per cent of those tried before Judge D, whereas in the other district courts of the same circuit only 62.5 per cent lost

pellate jurisdiction to review by appeal the final decisions of the district courts of the United States. In other words the single court of patent appeals, so far as patent cases are concerned, would take the place of the ten separate circuit courts. A decision rendered by this court would hold throughout the entire United States. This is one of the strongest arguments in favor of the single court. Another argument is that the judges would be selected on the basis of both engineering and legal ability, most of them, probably, from the patent profession. In addition to the judges, the court would have scientific advisers who would devote their full time to the court. Under the present system the judges in the several federal courts are faced with making patent decisions notwithstanding their lack of training in engineering matters. The following quotation taken from a patent decision is illustrative of this point:

> "I cannot stop without calling attention to the extraordinary condition of the law which makes it possible for a man without any knowledge of even the rudiments of chemistry to pass upon such questions as these. The inordinate expense of time is the least of the resulting evils, for only a trained chemist is really capable of passing upon such facts, e.g., in this case, the chemical character of Von Furth's so-called 'zinc compound', or the presence of inactive organic substances . . . .
>
> "How long we shall continue to blunder along without the aid of unpartisan and authoritative scientific assistance in the administration of justice, no one knows; but all fair persons not conventionalized by provincial legal habits of mind ought, I should think, to unite to effect some such advance".

The foregoing quotation was made in 1911. One reason why nothing has been done by way of new legislation to establish the proposed single court of patent appeals is that while the judges of such a court would be well qualified to understand engineering, it is feared that they would become too technical and narrow and lose sight of the legal aspect of the cases. It is a question whether it is better to have judges who are strong in law but weak in engineering as under the present system, or who are weak in law and strong in engineering as might be the case under the proposed single court of patent appeals. If it were certain that the judges would not acquire an over-technical attitude, but keep a well-balanced mind between law and engineering, then the court would be an excellent answer to the needed change in our patent system. Ideal as a single patent court might seem, however, there apparently is a feeling among legislators and others that judges should not be specialists in the sense of being expert engineers. It is argued that judges should keep tuned to the growth of general law lest they forget that patent law is law at all. Should the proposed single patent court develop into a narrow-minded tribunal, its effect upon the patent system would be no less damaging than that of a bureaucratic institution where power is concentrated.

System Compared with Principle of States Rights

Accordingly the distribution of patent cases to the ten circuits, as under the present system, safeguards our democratic principle of operation and may be likened to the principle of states rights. Should one circuit become hostile to patents, there always is the possibility that the patent owner may obtain service for starting his law suit in another circuit which is more favorable. *Fig.* 3 shows, for a three-year period, the percentages of patent cases lost by the patentee in the ten circuits. When the patentee loses, the court may hold the patent invalid, not

infringed, or both, and the chart is made up on this basis. The points on the full line curve show the percentage of patents lost in the circuit courts of appeals, the points on the dotted line curve showing the percentage lost in the district courts. It can be seen that the average lost in the circuit courts of appeals is approximately 82 per cent and in the district courts approximately 68 per cent. In other words, the average chance of a patentee winning upon his patent in the circuit courts of appeals is 18 per cent and in the district courts is 32 per cent. In the Supreme Court the percentage of patents lost is still higher than in the circuit courts of appeals.

Odds Are Against Patentee in Some Courts

The chart of *Fig.* 4 shows the disposition of patents involved in cases tried by the district judges for the 7th circuit, over a period of three years. In these charts the patents actually were tried and large sums of money were spent. Patents which were settled between the lawyers are not included. In the minds of the opposing patent attorneys who tried these cases, each side being equally confident of prevailing over the other, there existed a 50-50 chance of proving or disproving the invalidity or noninfringement of the patents. Referring to *Fig.* 4, Judge D decided cases involving 85 patents. The plaintiffs lost 67 and won 18, making a "lost" percentage of 78. The remaining nine judges decided cases involving 40 patents. In these cases the plaintiffs lost 25 and won 15, making a "lost" percentage of 62.5 which is somewhat closer to the 50-50 estimate in the minds of the patent attorneys. Judge D in effect constitutes a single court by himself.

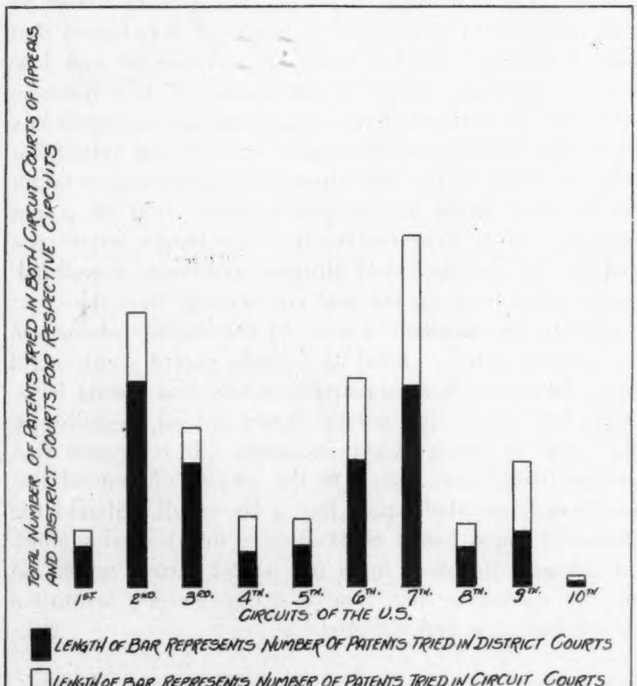


Fig. 5—Of the ten circuits, four had cases involving 653 patents while the remaining six handled only 219 such cases, within a period of three years

Most of the patents are tried in the 2nd, 3rd, 6th and 7th circuits, as *Fig.* 5 shows. These four circuits, including both the circuit courts of appeals and the district courts, had cases involving 653 patents in a period of three years, whereas the six remaining circuits had cases involving 219 patents.

When a patent is issued by the Patent Office, it is presumed to be valid and the burden is on the defendant to prove that the patent is invalid. This presumption grows out of the fact that the Patent Office has made a thorough examination of all the prior art patents and found that the claims in the application covering the new device possess patentable invention. Of the applications filed in the Patent Office, records show that in the neighborhood of 30 to 40 per cent are rejected or become abandoned. Thus the Patent Office weeds out approximately one-third of all the applications filed. When the patent is later sued on in court, the trial judge is again called upon to determine if the claims covering the new device define patentable invention. In other words, the trial judge must review the same question that the Patent Office previously passed upon, the trial judge, however, taking into account any additional prior patents which the defendant might have presented to the court as having a particular bearing upon the question of patentable invention.

Why Trial Judges Tend To Favor Defendant

In making his decision, the trial judge is not unmindful of the possibility of being reversed by the circuit court of appeals, inasmuch as the judges there hold more strongly against the patentee than the district judges, as shown in *Fig.* 3. In other words, the trial judge finds himself between two opposing factors, namely, the presumption that the patent is valid as issued by the Patent Office, and the recognition that the circuit court of appeals will likely reverse him if he should hold the patent valid. No trial judge likes to be repeatedly reversed. Accordingly, at the outset of the trial the judge is caused to lean on the side of holding the patent invalid, although secretly he might prefer to uphold the patent. As a consequence, the presumption of validity means nothing, and the defendant is relieved of his burden to prove invalidity. In fact, some district judges fear the appellate courts so much that they even place the burden on the patentee to prove patentable invention. The following conversation taken from a court transcript between the judge and a patent attorney, illustrates the predicament in which most trial judges find themselves:

> THE COURT: When the disposition against all monopoly is evidenced in the destruction even of the monopoly granted by the Government, the so-called presumption of validity is not a very weighty presumption; and though the burden is upon a defendant in respect to a prior use to establish such prior use by clear and convincing evidence—some of the old cases go so far as to say beyond a reasonable doubt—the burden is still on the defendant; and, likewise, the burden is on the defendant in respect to a defense of lack of novelty or lack of operativeness, that is to say, nonutility.
>
> But when you come to the matter of invention,

(*Concluded on Page* 172)

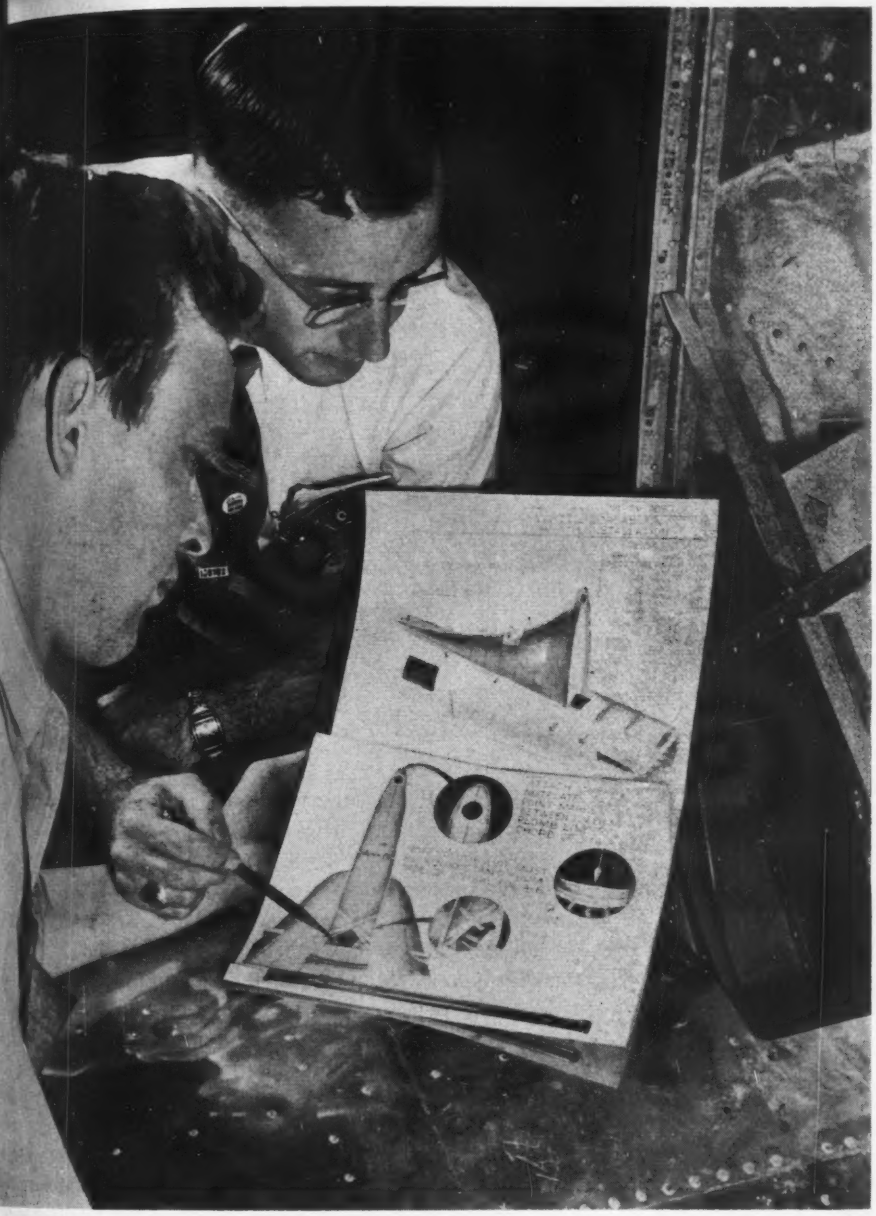


Fig. 1—An aid to production, photographs of perspective drawings enable the worker to visualize each step from beginning to final assembly

Photographs Aid Presentation of Design Details

By John W. Greve

POSSIBILITIES for use of photographs in the engineering department usually are overlooked or discarded as being too involved, too exacting or too limited to be practical. There are, however, many ways in which photography can be applied to conserve both drafting time and assembly time. Assembly time particularly can be saved through the easily understandable medium of photographs for training shop personnel which often is unaccustomed to line drawings. Because of present conditions involving rapid turnover of employees, teaching them to read line drawings is impracticable. Instead photographs of assemblies or photographs of perspective or pictorial drawings are finding widespread use.

When perspectives are required and models are available, photographs may be utilized, obviating the preparation of expensive perspective drawings. Even when drawings are desired because of the inability of photographs to show the proper view or internal parts, photographs of the assembled parts are of distinct help in

Fig. 2 — Right — Combination rendering and retouching shows assembly and operational details. Cutaway parts are rendered, balance of illustration is a retouched photo

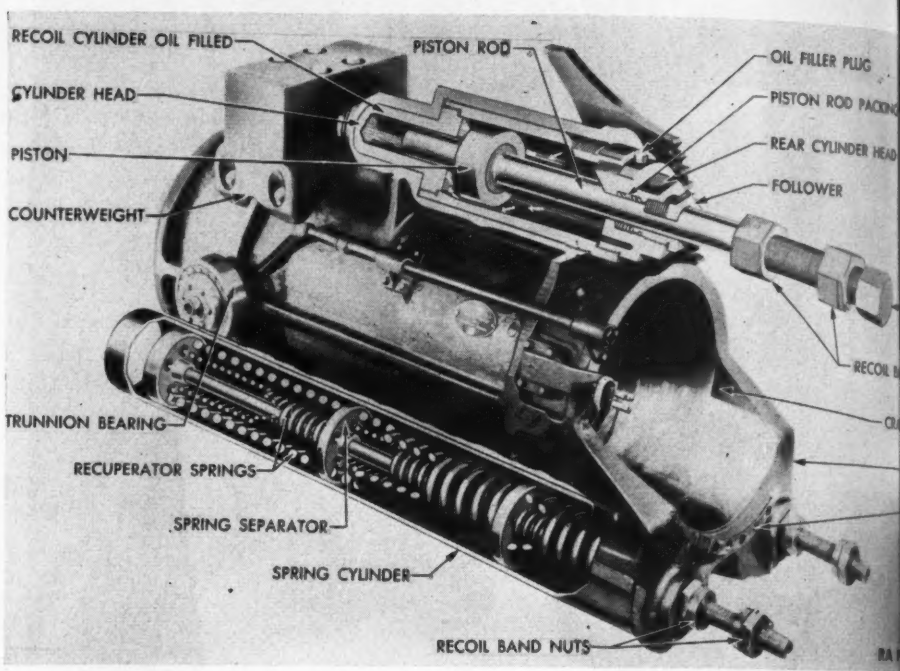


Fig. 3—Below—Exploded perspective is a rendering of carbon pencil on tissue

assisting—in this case—the draftsman.

The familiar type of illustrations used in automotive instruction books has been applied with improved techniques particularly by the Ordnance Department for preparing maintenance manuals. The same types of photography and art work have tremendous possibilities in preparing assembly instructions for the shop and are being employed to advantage by a number of companies, especially in the aircraft industry, *Fig.* 1.

Because photographs show only the exterior from one viewpoint, means must be found to circumvent this handicap. If the parts are too complicated to show the necessary details from a judicious selection of perspective the desired effects may be achieved by superimposing photographs, combining a photograph with a drawing, *Fig.* 2, use of exploded views, *Fig.* 3, or other expedients to effect the proper illustration.

In the use of photography it is always of primary importance to obtain a good quality illustration with proper attention being paid to lighting detail, contrast, perspective, separation of planes, separation of tones, proper background, elimination of irrelevant matter, etc. The time and money saved by proper attention to details at the time of taking photographs usually repays a hundredfold in time saved and in reduced retouching cost. Poor photographs are always expensive. As to retouching, it should always be handled to retain the photographic qualities as well as proper appearance of materials and finishes, *Fig.* 4. Planes should be handled in such a way as to give distinct variation.

Painted Parts Accent Photos

Photographing new machine parts usually produces excellent details when proper attention is paid to the primary rules of photography. Painting the assemblies which are desired to be illustrated often proves advantageous. In this connection excellent results are obtained by using a light battleship gray without any varnish so that no lustrous highlights are photographed to obscure detail. Proper photographic exposure is made for the part so painted so that the other parts show in relation but in subdued tones. For example, to illustrate a spring mounted in a chassis, the spring could be painted to subdue the frame in the photograph, giving much the same effect as the rendering in *Fig.* 5. Often it is desirable to show other parts in secondary relation to the primary part but not as subdued as the overall outline. In such cases darker tones of battleship gray are employed, being careful again to use a paint without luster.

As in mechanical drawings, photographs are best when a minimum amount of detail, consistent with portraying the desired information, is shown. In this respect several photographs, each depicting one specific element, produce superior results to one wherein many confusing details are shown. A further advantage is that parts may be removed in subsequent photographs of a series to show the operational relation of the assembly desired.

Phantom views usually offer the best possibilities for showing relation of operating parts in a machine, limit of motion relative to a housing or enclosure, etc. For instance, limiting movements of a part can be shown in phantom—the neutral position having full exposure and the limiting position half exposure. In this connection in-

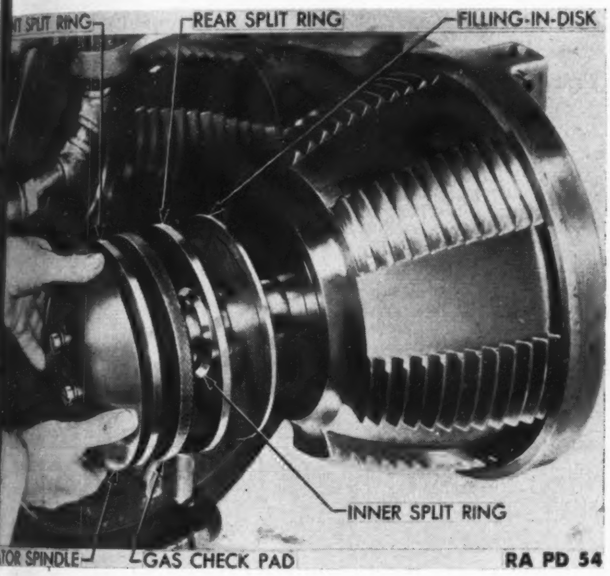


Fig. 4—Retouching retains photographic quality and natural appearance of materials

genuity on the part of the photographer is often needed to avoid ghosts (undesirable parts from the background showing through). If the background can be selected as a black body there is no trouble from this source. Photographs of artists renderings, Figs. 5 and 6, are employed to advantage to show phantom views.

Photography also may be used to show exploded views but skill is required in laying out the parts in exact relation and perspective without undue distortion so that the assembled positions of the parts are obvious. One method employed successfully is to use glass to support the part in the proper three-dimensional perspective. The glass is cut to necessary size and built up to the proper level for desired perspective by using small squares piled one upon the other. These stacks of squares supply an excellent adjustable support without tendency of slippage if properly arranged. Use of glass in this way does not cut out views of parts underneath. Careful selection of lighting obviates undue reflection. Reflections, however, are not often of any consequence because views of this kind must have the background carefully opaqued to obtain the desired results.

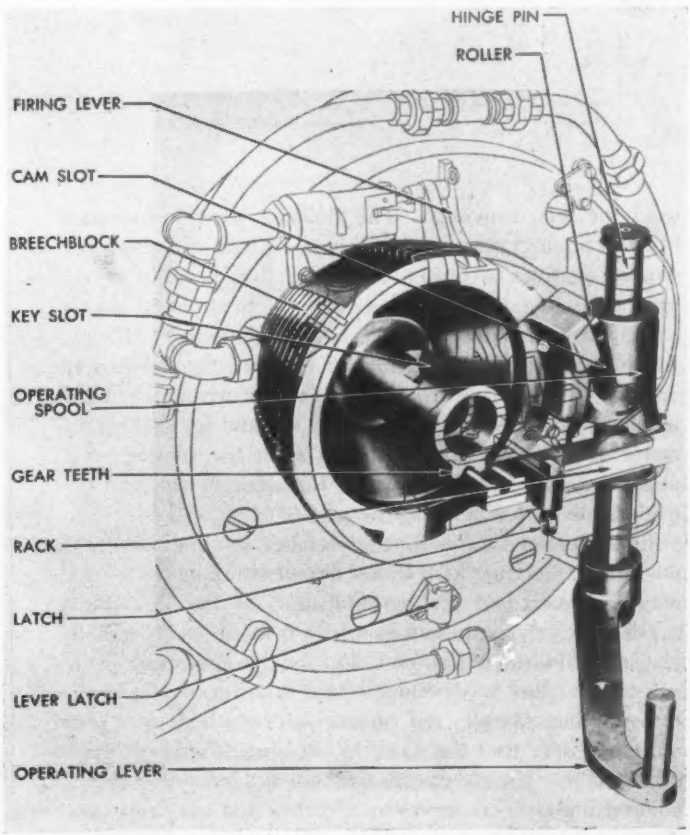
Another method involves the utilization of a large piece of plywood with holes drilled ½-inch apart with suitable iron rods inserted in these holes to support the parts in their proper perspective. A third method utilizes an ingenious device called the Schrader universal jig. These methods, however, do not have the advantage of the glass supports in that back lighting cannot be used. Wires, putty, beeswax, pins and other props for positioning parts, are useful in making exploded views.

Relation of a detailed assembly to its machine or housing may be shown by airbrushing or otherwise subduing the general outline so that the detail part is made conspicuous by the contrast. This procedure may often be used to advantage to show combinations of assemblies using the same background for outline photographs. When this is done it is necessary to measure the perspective views carefully so that superimposed photographs will appear in proper relation. Such superposition has advantages when part of the frame would otherwise obscure details of the assembly. Pictures of this nature are best taken at the same time so that superposition will fit exactly and measurements can be easily reproduced.

Cutaways make clear the relations of inner parts and may be produced by photographing (1) actual cutaway, (2) drawings prepared from blueprints, *Fig.* 5, (3) combination drawing and photograph, *Fig.* 2 and (4) tone wash drawing prepared from mechanical drawing. Drawings prepared from blueprints may have tones added by a retoucher to make a wash drawing which appears photographic.

In reproducing illustrations for use in the shop or else-

Fig. 5—Below—Airbrush rendering shows the relations between parts in phantom view



where, the problem is not simple. There is no substitute for a good reproduction. If quantities are small, the prepared illustration with art work is photographed and contact prints are used. Also photostats may be utilized. If quantities are larger, say 50 to 100, it may be advisable to make a negative suitable for printing in a blueprint or black and white type of machine. Larger quantities are usually reproduced by the offset or letter-press method.

Preparing suitable masters for reproduction in contact printing machines such as used for engineering line drawings has not been highly successful where good fidelity is desired. There are two reasons for this: The sensitized paper used is high-contrast without sufficient latitude for halftone illustrations, and operators (experienced in producing line-drawing reproductions) are unfamiliar with the precise exposure required for halftone work.

Satisfactory results have been obtained by "screening" a photograph to produce a negative with 85 lines to the inch which could be printed with an exposure close to that used for line drawings. The finished print looks much like the photographs in newspapers. Linen negatives and paper negatives are often employed, but because of the photographic emulsions on one side the heat of the printer causes objectionable curling, making handling difficult. In case a positive printing machine is used the transparent original is made positive instead of negative. In transferring a negative to linen, detail is lost and for that reason many users prefer to print direct from the photographic negative. Also, because of the thickness of the film objectionable curl is not experienced.

Reproductions of wiring diagrams or line drawings where color coding of various circuits or lines is desired may be performed photographically. If the line drawing is relatively simple an exposure is made on a specially prepared film having an emulsion for dye inhibiting printing. This film is developed and transfer dyes—black, brown, blue, orange, red, green—are brushed on. It is only necessary that the dyes do not touch the lines of another color. Excess dye is washed off, leaving only the colored lines. This serves as a matrix and may be transferred to almost any photographic paper. As many as 75 prints may be made from one matrix. If line separation is complicated, separate matrixes may be made for each color, either by filtering or by separation line drawings, and transferred in register to the printing paper.

Another method of producing colored or coded line diagrams involves the use of a direct color-printing paper. There are three ways in which this may be accomplished. One is to expose a black-line negative and dye the various transparent lines corresponding to the colors desired. No filters are needed to print this negative on the color paper in one operation. Another involves printing separation negatives through filters complementary to the color desired. The third utilizes three-color separation negatives exposed from a colored original. These negatives are printed with their corresponding filters.

Photography can also be useful in providing templates to scale of outlines for standard parts frequently used on drawings. Assembled units as well as bolt heads, screw threads, angle pieces, sections, etc., are typical. Such parts are drawn accurately and then photographs produced for various scales that might be employed, making one drawing serve for several templates.

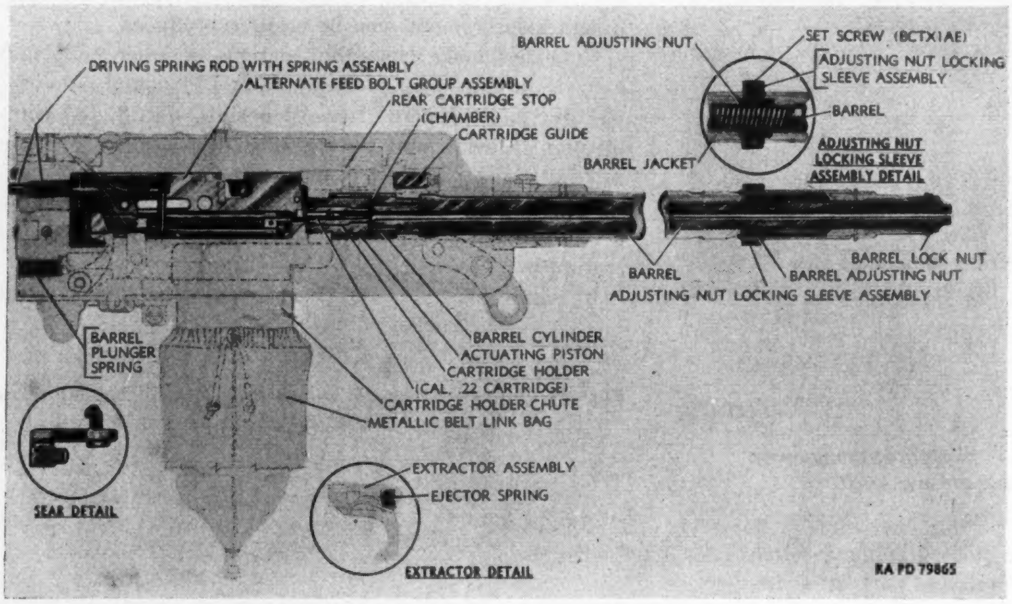


Fig. 6—Carbon pencil rendering of machine-gun trainer shows relation of parts, with housing in phantom

Microfilming of drawings for record purposes and insurance against loss of the original is another useful adaptation of photography. Photographing of pencil drawings to preserve the original, simplify drafting for design changes and assure better prints is discussed in a previous article "Duplicate Tracing Minimizes Drafting Work" in the November 1943, issue.

Cooperation of the following in supplying information and illustrations for use in this article is greatly appreciated: Ansco division of General Aniline & Film Corp., Binghamton, N. Y.; Bell Aircraft Corp., Buffalo; Boeing Aircraft Co., Seattle, Fig. 1; Charles Bruning Co. Inc., Chicago; Consolidated Vultee Aircraft Corp., Vultee Field, Calif.; Defender Photo Supply Co. Inc., Rochester, N. Y.; Douglas Aircraft Co. Inc., Santa Monica, Calif.; General Electric Co., Schenectady, N. Y.; The Haloid Co., Rochester, N. Y.; War Department, Office of Chief of Ordnance, Washington, D. C., Figs. 2, 3, 4, 5, 6; and Gus Wetzler, Commercial Photographer, Cleveland.

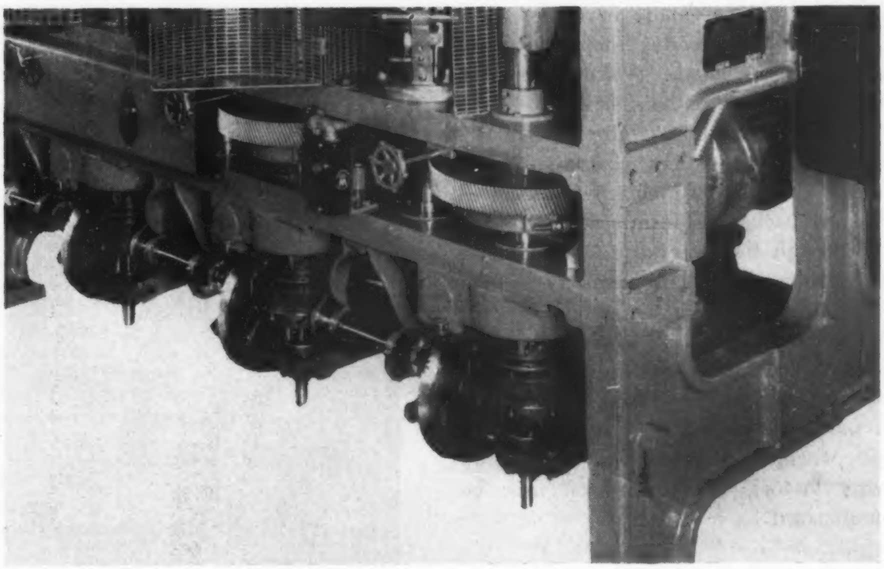


Fig. 1—Six-spindle wire-insulating machine employs disk-type friction clutches to make possible smooth gradual speedup. Partial view shows clutches at lower ends of three of the six vertical shafts

Specifying Intermediate Components for Machine Drives

FIRST of a series giving designers the latest information on clutches, brakes and couplings. Basic operating principles and characteristics are discussed in a manner gaged to aid in evaluating the different types of units for specific applications

By Richard K. Lotz

Part I—Clutches

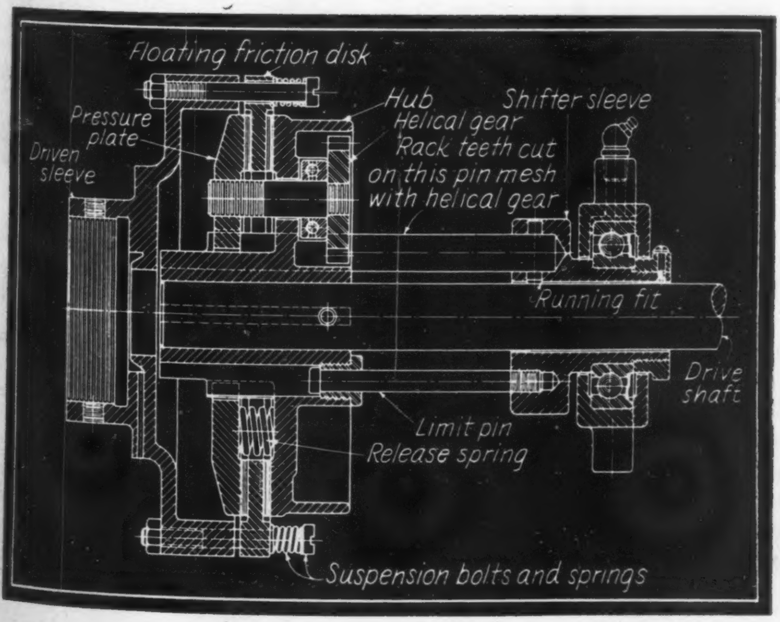


Fig. 2—Mounted vertically (right end down), this friction clutch incorporates friction disk which floats on springs

IT HAS been a matter of considerable importance that progress in the design of clutches proceed apace with the many improvements which have been effected—especially during the past few years—in the design of the mechanisms which clutches serve.

There was a time when the machine designer was required to create veritably every part which went into his machine. This of course meant designing any clutches required as well as parts and units such as gears, speed-reduction units, brakes, couplings, etc. Fortunately, and this is undoubtedly one of the prime contributing factors to the strides

which have been made in design during the past decade, many machine units now are made by concerns that are specialists in their fields. Thus there are few clutch requirements today which cannot be met competently by one of the standard clutch units available in "packaged" form.

Although these units are available to the designer, it still is primarily his job to *select* and *specify* the best clutch for his specific application. Time spent in carefully analyzing the requirements of his machine will pay big dividends in its performance and life. The type and size of clutch selected will depend on many factors, most of which may be summarized as follows:

1. Required torque capacity
2. Space available
3. Permissible shock during engagement
4. Type of control (manual or automatic)
5. Necessity for "jogging" or "inching"
6. Effect of clutch weight on inertia
7. Balancing required (high-speed applications in particular)
8. Ease of maintenance
9. Frequency of engagement
10. Expected or required life
11. Heat dissipation.

In determining the amount of torque which will have to be handled by a clutch, it is important to take into account not only the theoretical calculated steady torque but the fluctuations in torque which may be present during the operation of the driven unit. Intermittent momentary rises in torque exist in many machines and often cannot be calculated readily. In these cases they must be estimated as closely as possible or determined by actual test. In any case, where quantities of machines are to be built, it is of course wise to select the clutch only after suitable tests have proved it satisfactory for the application.

Space Requirements

Power-transmitting ability often can be packed into a surprisingly small clutch. Witness for example the really compact multiple-disk clutches and the roller-type overrunning and indexing clutches. Nevertheless there always is a minimum size to which any clutch can be made and if this minimum is being depended upon in determining space allotments for the clutch and the surrounding mechanism, it is best to be doubly sure that the clutch will handle its loading adequately. Should it prove to have insufficient torque capacity, correcting the condition will entail employment of a larger clutch which in turn will necessitate altering the surrounding mechanism to provide the additional space. While it might of course be possible to make the smaller clutch transmit the power by forcing it, this rarely is advisable since it results in excessive wear and involves repeated inadequate replacements.

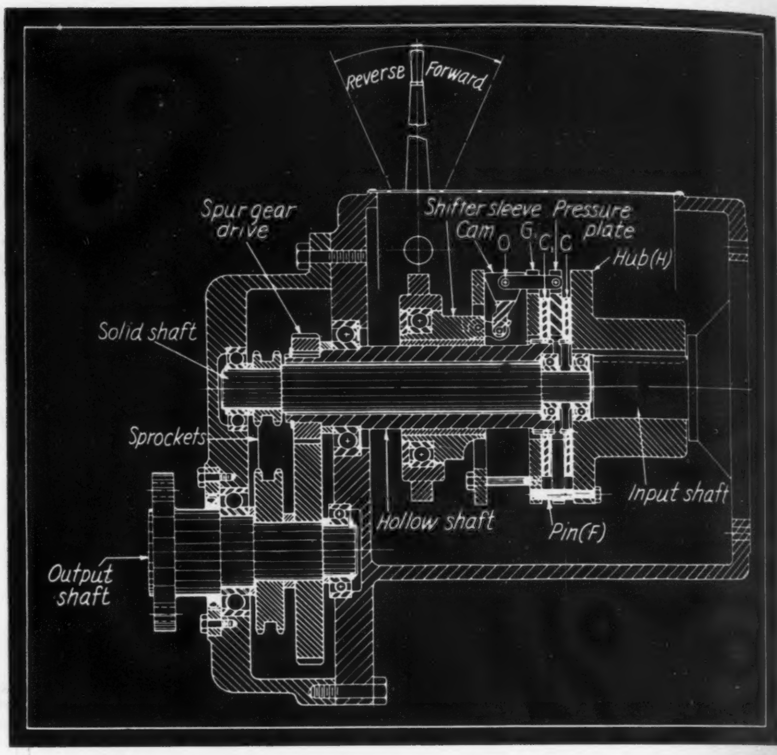
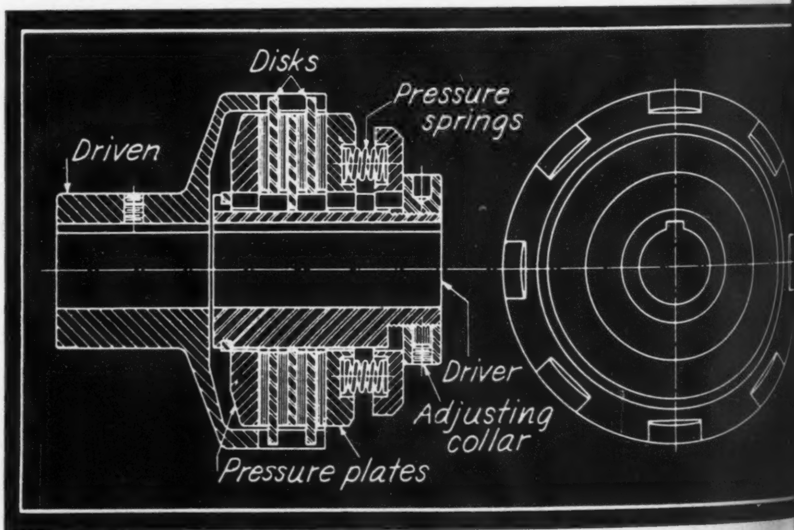


Fig. 3—Above—By selectively driving either the solid or hollow shaft, two-disk friction clutch makes possible simple reversing drive. With spur gears replacing sprockets, a two-speed one-way drive is produced

Fig. 4—Below—Slip clutch employs spring-loaded friction disks. These units transmit normal loads positively but slip when subjected to overloads



In most cases it is necessary to bring a driven mechanism up to the speed of the driver with only as much shock as is demanded by the operating requirements of the machine. Applied at a rate commensurate with the time permitted for engagement as well as with the shock loading which the machine will tolerate, friction provides a convenient and readily controlled means of coupling and is in fact the means utilized for transmitting power in the majority of clutches in use today.

Obviously there would be substantially less need for the friction type

of clutch if abrupt starting with its attendant shock were not a vital factor in the design of machine drives. The amount by which shock can be reduced by the simple expedient of permitting a gradual speed-up over a period of seconds is considerable. Starting never should be any more abrupt than is absolutely necessary. Where accurately-cycled mechanisms demand starting free of slippage, efforts might well be made to dampen the shock through the use of spring-loaded coupling devices having the faculty of returning the affected members to accurate register after absorbing the initial shock. Other devices, similar in character, also may merit the careful consideration of designers.

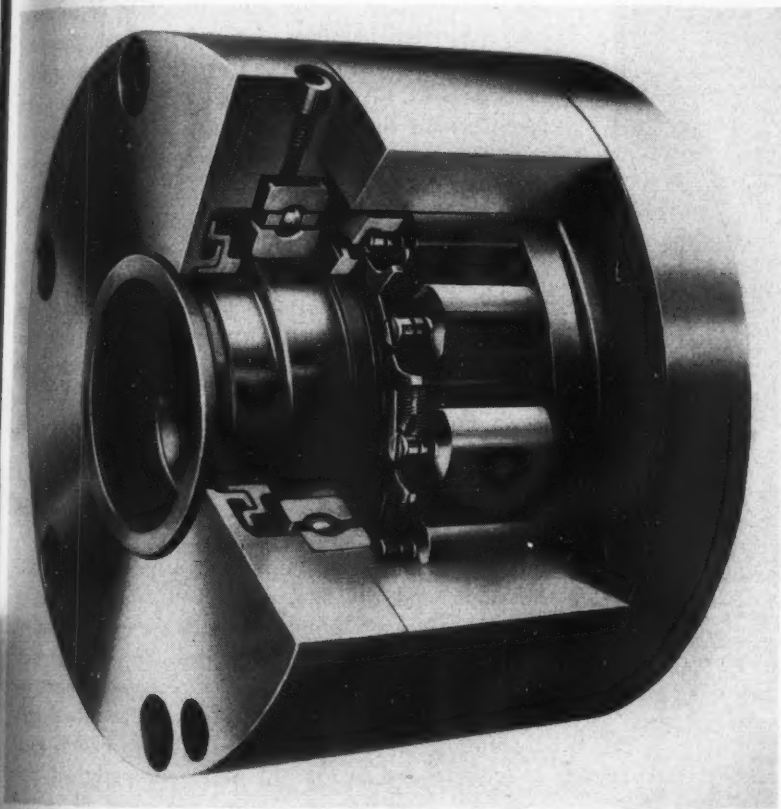
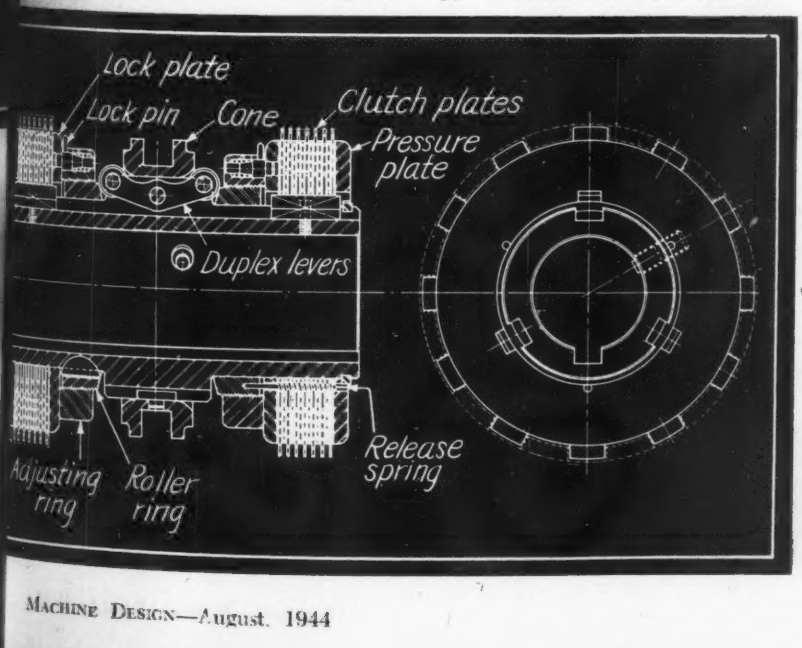


Fig. 5—Above—When engaged, the series of cylindrical cams in this overrunning clutch wedge between the inner and outer members of the clutch. Such clutches drive positively in one direction, turn freely in the other

Fig. 6—Below—Multiple-disk clutches pack a great deal of torque-transmitting ability into a small space. Type illustrated is double-ended



An application which exemplifies employment of gradual, smooth speed-up through the use of a friction clutch is the six-spindle wire-insulating machine made by the American Insulating company of Philadelphia. In this machine, part of which is shown in *Fig.* 1, each of the six vertical spindles (three are shown) drives sets of bobbins and flyers for applying the insulation to the wire. After the wires and insulation are threaded properly it is essential that the revolving parts, which have compound motion, be accelerated with extreme smoothness; otherwise either the wire or insulation may break, necessitating rethreading.

It was required in this case that a manually-controlled clutch be employed which would permit the operator to pick up the load with the requisite smoothness. This resulted in the adoption of the unit shown in *Fig.* 2. Pressure is applied through the rack-and-pinion type of mechanism which is nonoverhauling and has no definite point of engagement.

Floating Friction Plates Employed

Since these clutches are applied vertically, a special design was needed in order to make the friction plate clear the pressure plates in the release position, thus preventing any drag from continuing to revolve the spindle. The friction disk is driven by special studs which have springs on them to support the weight of the disk and cause it to float in the proper position clear of both pressure plates when the clutch is released. The ball-bearing type of shifter collar on this clutch was mounted downward and the threaded end of the short clutch sleeve was coupled to the spindle drive.

A simple way to obtain either a reversing or two-speed, one-way drive by purely mechanical means is illustrated in *Fig.* 3. A customary type of disk friction clutch is used, having two disks, each of which has an internal gear cut on its inside diameter. When the shifter sleeve is moved toward the clutch proper, the cams, of which there are three equally spaced, pivot on their pins *O*, causing them to push against the hardened plate and thus clamp disk *C* between the pressure plate and the hub *H*. The hub, being keyed to the input shaft, drives the solid shaft through the gear which is in mesh with the internal gear of the disk *C*. Thus the output shaft is

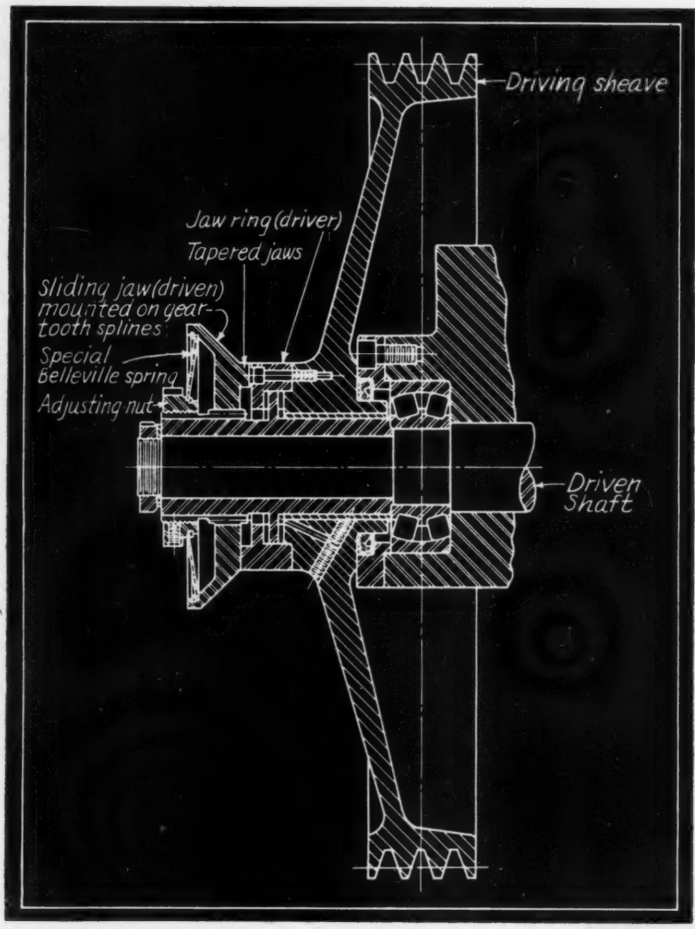
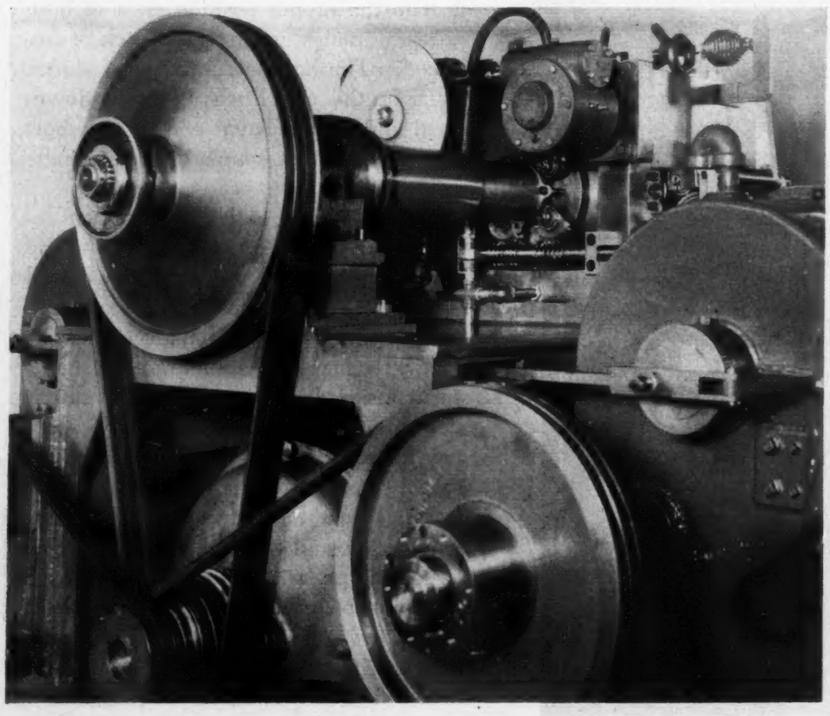


Fig. 7—Above—Overload release clutch of the "snap-out" type utilizes tapered jaws loaded by Belleville spring. This clutch will transmit torque up to its peak without slippage. Subjected to overloading, however, it disengages quickly and completely

Fig. 8—Below—Typical application of the overload release clutch shown in Fig. 7 is on the drive of this Sendzimir strip-rolling mill the action of which is sensitive to overloading



driven by means of the sprockets.

When the shifter sleeve is moved away from the clutch proper, the cams—pivoting about their pins—push against plate G_1 and pull the pressure plate by means of the linkage to clamp the friction disk C_1 between the pressure plate and plate G_1. In this position the drive is from the hub H, through pins F, through disk C_1 to the gear cut integrally on the hollow shaft, through the spur gears and to the output shaft.

It will be seen that both the solid and hollow shafts are driven in the same direction of rotation and it is what happens in the drive from them to the output shaft that makes the unit either reversible or two-speed. As shown, when driven through the sprockets the direction of rotation will of course be opposite to that obtained when the drive is by means of the spur gears. By substituting spur gears for the sprockets at a ratio different than that now employed from the hollow shaft, a two-speed one-way drive is obtained. The function of the clutch in this application is to disengage fully either of the shafts while the other is driving or to cut out the drive entirely by moving the shifting lever to neutral.

Protecting from Overloads and Shocks

Many times a mechanism must be protected against overloading or shock loading which might occur due to jamming, mistakes of the operator, etc. Stripped gear teeth, bent shafting, cracked housings, and the like, can result from failure to provide adequate protection and a simple solution is often effected through the use of a shear pin. However, shear pins require replacement and replacements take time. At slightly more initial cost a better solution is provided by a slip clutch such as is illustrated in *Fig.* 4. When properly set, this unit insures a positive drive at all normal loadings, but the instant an overload occurs, slippage takes place. Thus it effectively replaces the shear pin and offers the advantage of requiring no replacement. The unit shown in the illustration is essentially a simple spring-loaded friction clutch provided with an adjustment so that the torque at which it will slip can be varied to suit the conditions of the particular application.

Similar to the immediate foregoing is the application that requires positive drive right up to the point of overload and then absolute cut-off of the drive. Such a condition exists in the Sendzimir strip-rolling mill, the drive section of which is shown in *Fig.* 8. This mill turns out metal strip of exacting tolerances and extremely fine finish. Overloads on its rolls would distort them and render the finished strip less accurate. To meet the requirements of this job the overload release clutch, *Fig.* 7, was developed.

This clutch will carry any torque from zero up to the maximum for which it is

adjusted without the slightest tendency to disengage, but once the torque value for which it is set is reached, instant and complete disengagement takes place. The Belleville spring which is the controlling element in this clutch is of the "snap-over-center" type. When deflected to the flat position, its load capacity drops to zero and just beyond the point at which it is flattened it instantly turns "inside out" and pulls the jaw-clutch part of the unit into the clear, completely severing the connection that carries the drive. In order to re-engage the clutch, it is necessary simply to force the sliding-jaw ring back into position after approximately lining up the jaw members. Although it has not been done in this particular application, other applications have been equipped with a re-engagement control that can be operated pneumatically from a remote station.

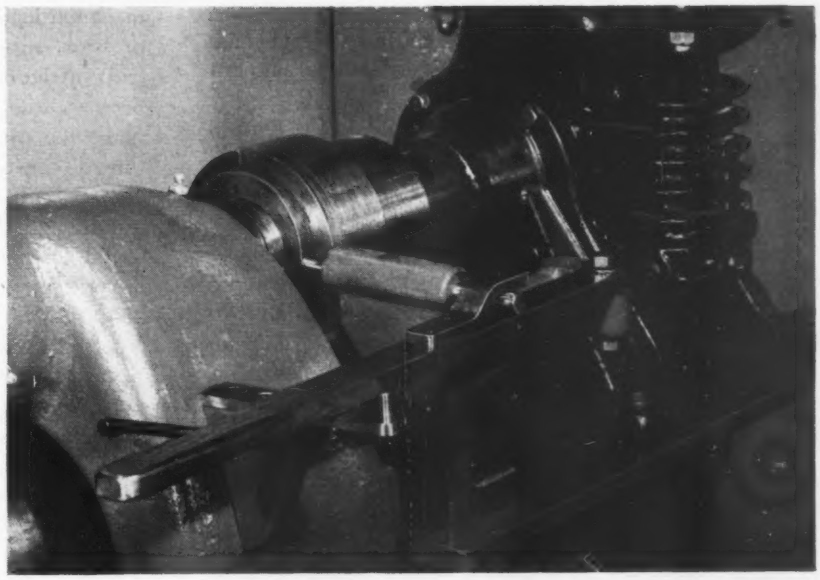
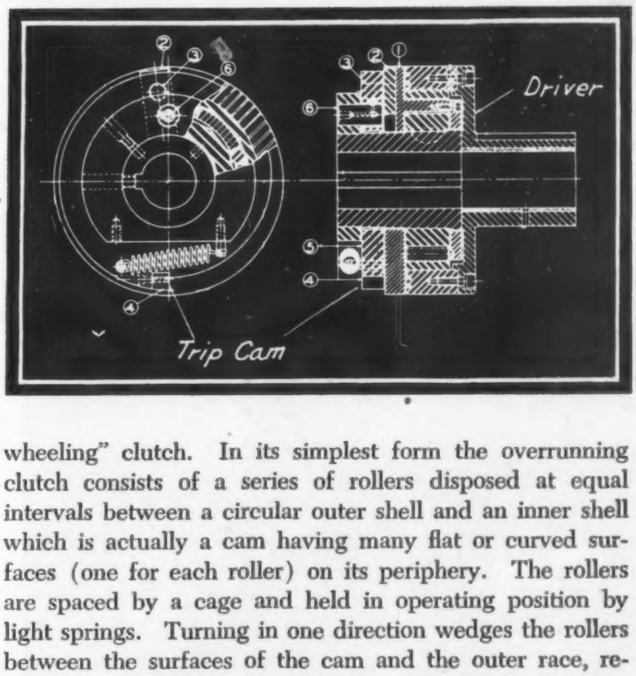


Fig. 9—Above—Partial view of shell-marking machine showing single-revolution clutch on main spindle and trip pin with its spring-loaded hand lever-control

Fig. 10—Below—Single-revolution clutch combines roller-type overrunning clutch with trip-cam release mechanism



It sometimes is found that an application which demands protection against more than a certain amount of torque after the machine is in operation, must be able to carry more than that amount during the starting and accelerating period of the machine. In such cases, the remote-control operation of these overload-release clutches can be so arranged that the operator can turn the control on during starting and off after the machine has reached its operating speed.

Multiple-Disk Types Are Powerful, Compact

Referring back briefly to the disk type of friction clutch, it would be difficult to find a unit that packs more torque-transmitting ability into a small space than does the multiple-disk clutch shown in *Fig.* 6. The model illustrated is double-ended and finds wide application in automatic reversing drives. Other applications utilize the disks at one end for clutching and the disks at the other end for braking. Single-ended models of these clutches also are available.

Various makes of these sturdy, compact units are regularly produced, the primary differences between them being in the mechanisms employed for engaging and disengaging, and in the shape of the driving disks. In the unit depicted in *Fig.* 6 the cone, actuated by a conventional bronze throwout collar, can be moved axially to right or left. In so doing it exerts pressure on the disks through the roller ring and the rollers which are carried by a series of duplex levers. Quick release when the cone is returned to its neutral position is effected by the release springs which push against the lock plate and roller ring to free the disks. Adjustment of pressure is made by pulling the spring-loaded lock pin out of engagement with the lock plate and turning the adjusting ring relative to the roller ring. When a clutch of this type is specified, particular care should be exercised in ascertaining that it has adequate torque capacity because, due to the inherent difficulty of dissipating heat where multiple disks are used, overloading can result in excessive heating.

One of the most interesting clutches is the overrunning type which sometimes is referred to as an "indexing, freewheeling" clutch. In its simplest form the overrunning clutch consists of a series of rollers disposed at equal intervals between a circular outer shell and an inner shell which is actually a cam having many flat or curved surfaces (one for each roller) on its periphery. The rollers are spaced by a cage and held in operating position by light springs. Turning in one direction wedges the rollers between the surfaces of the cam and the outer race, resulting in a positive nonslip drive. Turning in the opposite direction frees the rollers and the drive becomes freewheeling. Due to the pressure exerted by the roller springs the rollers are always in position for instantaneous engagement or disengagement.

Some of the more important applications for which overrunning clutches are particularly suitable are:

1. Intermittent feeds in which rollers, cams, gears, conveyors, etc., are actuated only when their drivers are turning in a given direction and cease moving when the direction of the driver is reversed
2. Driven mechanisms utilizing two prime movers (stand-by drives). Each prime mover is connected to the driven mechanism by an overrunning clutch with the result that either can cut in or out automatically

3. Booster drives in which the drive first is effected by one motor through the clutch and later picked up by another motor, the clutch automatically cutting the first motor out
4. Multiple-speed drives. Sometimes the overrunning clutch is used in conjunction with a disk-type friction clutch to obtain various automatic speed-ups and speed reductions
5. One-way drives which cannot be permitted to reverse. In such cases the clutch is used as a backstop.

Actually this listing of applications is not complete. Others, many of them ingenious, are in use but they all are predicated on the ability of the overrunning clutch to grab positively in one direction and turn freely in the other.

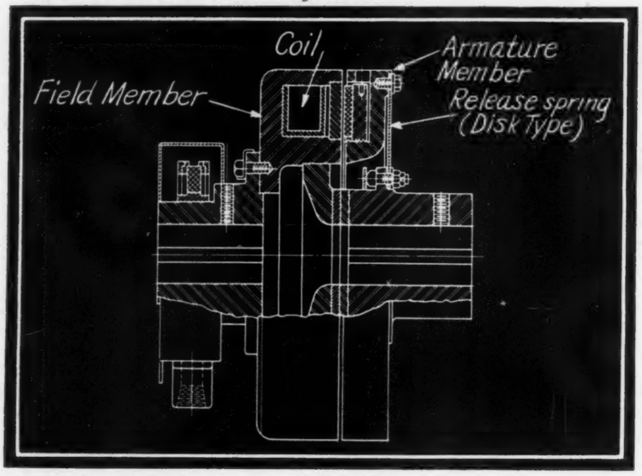


Fig. 11—Above—Friction type of magnetic clutch showing primary parts. Such units are ideal for remote control

Fig. 12—Below—Application of serrated (toothed engagement surfaces) magnetic clutch in equipment of steel mill

A type of overrunning clutch which has proved highly successful is pictured in *Fig.* 5. In this design the clutching action is produced by the series of cylindrical cams which, when driving, are wedged between the inner and outer clutch members. Actually, the curved surfaces of these cams are not true arcs but specially shaped contours designed to increase the locking force with increases of clutch loading. Each cam is actuated by a small compression spring through shouldered extensions at both ends of the cam. The gear-tooth mesh shown established between the cams and inner clutch member serves to space the cams, keep them in alignment, react the pressure of the cam springs, and keep the cams in constant contact with the outer member of the clutch.

Although the unit pictured in *Fig.* 10 generally is referred to as a single-revolution clutch, it can be used to provide any number of controlled revolutions. Essentially it is a roller-type clutch, similar to the overrunning type, with engagement and release effected by an auxiliary mechanism which consists of a spring-loaded trip cam, pivoted (on pin 6) at a radius from the clutch center and connected to the roller cage 1 through pin 3 and block 2.

As shown in the illustration the clutch is engaged, the trip cam being pulled in a counterclockwise direction by its spring and in turn urging the cage 1 in the same direction through pin 3 and block 2 which can slide radially in a groove cut in the face of the cage. (The end view at the left of the illustration shows the rollers wedged between the flats on the inner member and the circular race of the outer driving member.)

Trip Lever Releases Clutch

Release of this clutch is accomplished by a trip lever mounted adjacent to the clutch and in such position as to permit it to contact the trip-cam hardened insert 4. Immediately such contact is made the trip cam stops revolving, the driving portion of the clutch continues through the small fraction of a revolution required to push the cage and rollers to their release position, and only the driving member continues to turn.

Single-revolution clutches are used for cycling all types of intermittent drives. A typical application is the shell-marking machine, *Fig.* 9, made by James H. Matthews & Co. In this machine one revolution of the marking wheel (immediately above the end of the shell) is required for the marking operation on each shell. After the clutch is engaged by means of a hand lever, the spring-loaded trip lever pushes on the trip pin which rides on the contour of the trip cam and effects the release.

Magnetic clutches, which of course are controlled electrically, offer automatic or remote manual control with quick positive action and do not require toggles, links, shifting levers, etc. Some of the applications to which magnetic clutches are well suited include their use as positive engagement couplings for synchronizing rolling-mill screwdown motors, flying shears, cutoff machines and uncoilers; as slip clutches to prevent mechanism damage under sudden loads and jams; as accelerating units on machines where the driving member must be permitted to start without load and then is required to pick up the load smoothly; for frequent starting and stopping of such equipment as compressors and accumulator pumps; and for intersectional locking of certain machines as employed in the paper industry.

A magnetic clutch, as shown in *Fig.* 11, essentially is comprised of two sections, namely the field member containing the coil, and the armature member. Current is

(*Concluded on Page* 174)

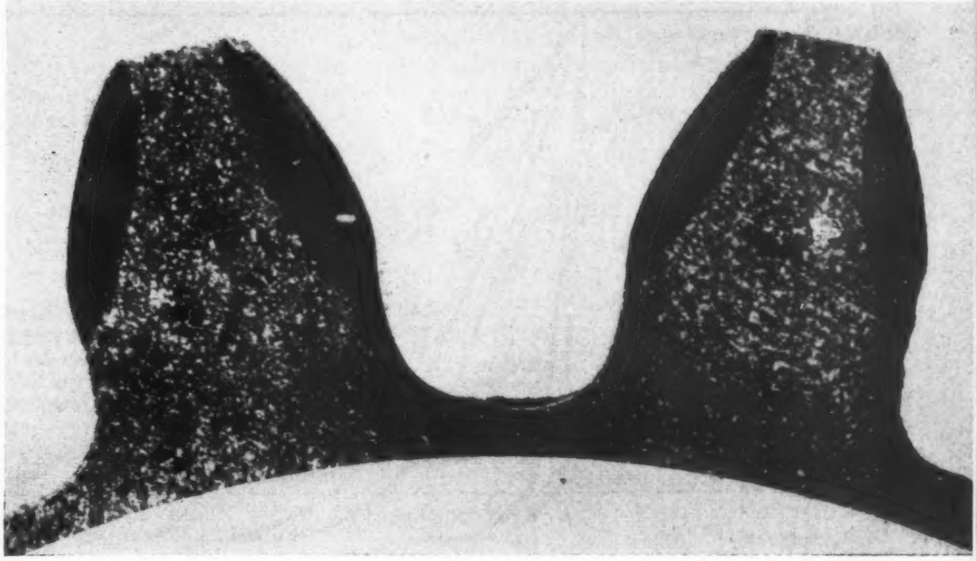


Fig. 1—Flame-hardened gear teeth showing case developed. Flame hardening does not alter the chemical composition of the material being treated

What New Heat Treatments of Steel Presage in Design

By Harry W. McQuaid
Republic Steel Corporation

GREAT progress has been made, under the pressure of wartime requirements, in developments of heat-treating processes for steel. Lashed by the unrelenting demands of the war, many of these developments have reached the stage of commercial application. Prominent among such recent developments are: (1) Induction and flame heating; (2) isothermal heat treating; (3) improved quenching; (4) shot blasting, etc.; (5) controlled atmosphere.

Strictly wartime developments might be limited to: (a) Use of multiple lower alloy steels to replace simpler steels of higher alloy content; (b) water quenching of alloy steel armament.

Hardening steel via induction heating has come into its own during these recent war years and, when its true nature along with its natural limitations are comprehended fully, it offers much to the designer. By using the proper induction coils and by controlling the frequency and power of the electric currents employed to give the required depth and speed of heating, it is possible to obtain heated areas to depths desired in a matter of seconds. Thus, it is a method ideally suited to mass production procedures.

Abstract of a paper presented at the recent national materials meeting of the Society of Automotive Engineers in Detroit.

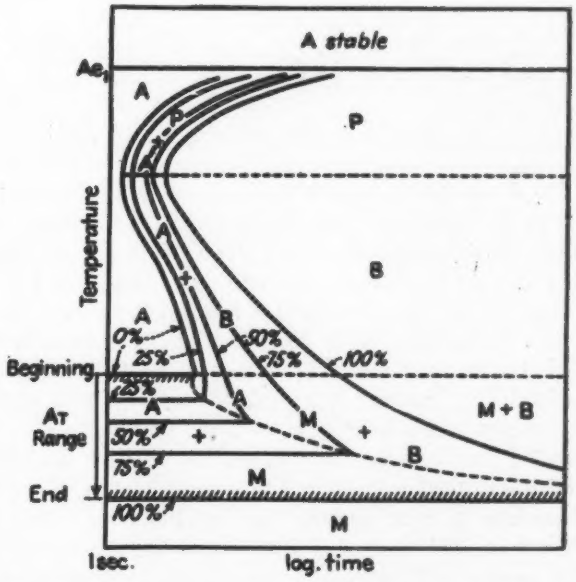


Fig. 2—Modified S curve. Full lines show per cent austenite transformed. A, austenite; P, pearlite; B, Bainite; M, martensite or martensite transformed

As indicated, induction heating has, of course, some metallurgical and mechanical limitations which must be considered. Properly worked out, however, it promises to give more leeway to the designer. In objects of simple symmetrical shape, such as pins and bushings, it permits the use of lower alloy and higher carbon combinations

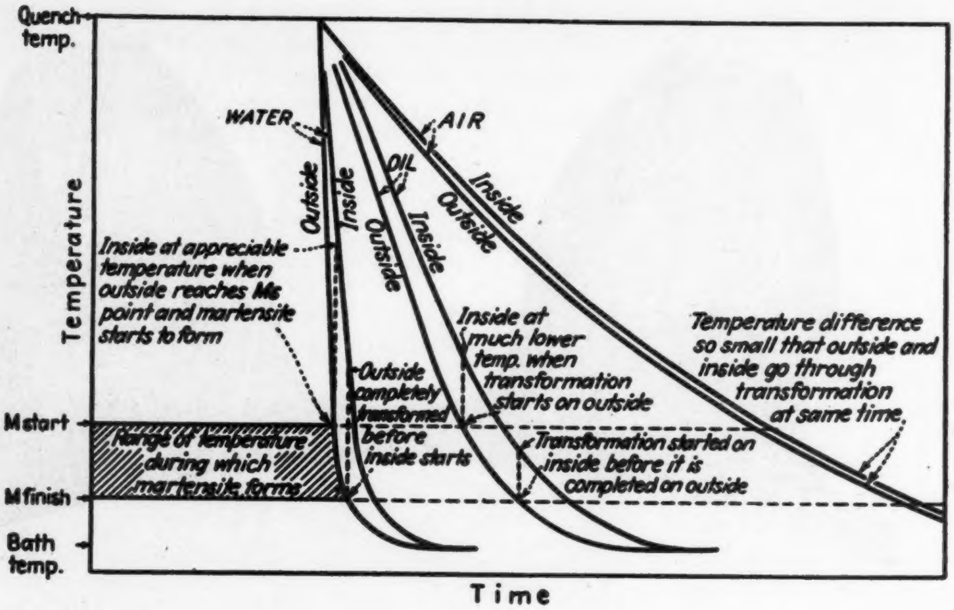


Fig. 3—Rates at which outside and inside of hardened piece cool when quenched in water, oil and air are shown by these curves (Shepherd)

which can be water quenched without excess distortion or internal stressing. This means higher hardness and greater load-carrying capacity. Primarily, the use of induction hardening should be limited to shallow-hardening steels, i.e., lower alloy steels, otherwise the severe quench may result in a highly stressed surface which, if it does not crack, has much decreased load-carrying capacity[1].

As its use and limitations become more familiar the application of induction heating is spreading rapidly. It can be used on internal as well as external surfaces and lends itself to "in line" applications. It can be effected automatically with simple pushbutton control and, where properly applied, provides a finished part which gives extremely satisfactory performance.

Effects of Induction Heating on Design

To the designer, the principal advantage of this method lies in its application to localized zones. This makes possible the designing of parts practically free from distortion due to hardening. In addition, parts can be designed so that the stresses in a finished piece, at a given point, can be relieved by local heating. Parts can be designed with only internal surfaces or projections subject to hardening, making it possible to make built-up welded or brazed assemblies prior to heat treating. Induction heating has valuable application in brazing operations and the distortion incidental to overall heating of welded assemblies in heat treating can be eliminated. The use of induction heating of gears by applying the heat to the rim section only permits wide latitude in built up webs and easily machined hubs.

Much that has been said about the use of induction heating can be said also about flame hardening (*Fig.* 1), although not all of the advantages offered by properly applied induction heating are available with flame hardening. However, for some applications there is little choice between the two methods. Every possibility should be studied carefully so that if flame hardening will satisfy, then the advantages present in much lower cost of installation should be obtained. In any event, an evaluation of the metallurgical and operating aspects of the two methods should be made by the designer before he decides on the design of the part or the heating method to be employed.

The advantages of induction heating in permitting localized softening of hardened parts should be familiar to every designer in connection with the attachment of hardened parts to other parts, in relief of stresses set up in fabrication, and in the possible forming of heat-treated parts. An example of the first case is found where a hardened flange is softened to permit drilling and tapping. An example of the second case is where induction heating is used to soften the bases of armor-piercing shot. For the last case, heating of the nose of a heat-treated high explosive shell to permit nosing-in is an excellent example.

Although isothermal heat treating was practised many years ago, it was in an undeveloped state until the advent of the constant temperature transformation curves of Bain and Davenport[2], about 1930. This work led to the development of constant-temperature transformation heat-treating practice on a commercial basis. This practice, because of limitations of size and analysis usually was confined to small sections and carbon or low alloy steels.

The publication of studies for many grades of alloy steels showing the time required at all temperatures for the transformation of the austenite to ferrite and the precipitation of the carbon in form of carbide to form martensite or other structures, led to application of this knowledge in commercial heat treating. Thus it was found that after the properly heated part had been cooled past the "knee" of the "S" curve without transformation (see *Fig.* 2) there was no great hurry in cooling, and parts could be straightened without decreasing hardenability greatly.

When the austenite transforms and carbon is precipitated to form the structure of maximum hardness known as martensite, the change is accompanied by a definite increase in volume. It is this increase in volume which contributes to most of the stresses set up in quenching. Since

[1] H. B. Osborn Jr., Vol. 79, *Transactions*, The Electrochemical Society, 1941; C. J. Holslag, *Welding Journal*, Feb., 1941; E. Blasker, *Symposium*, A.S.M., 1940.

[2] Davenport and Bain, Vol. 90, *Transactions*, A.I.M.E., 1930.

cooling proceeds from the surface to the center of the piece, if it is drastic—as in water—there always will be, at the time of center transformation, a considerable difference in temperature between the outside and center in water quenching. Thus, with the outside of the part cold and rigid and the center expanding in transforming, surface tension stresses will be set up which make any assumption of the designer extremely uncertain. This of course is true primarily of a deep-hardening steel. Any steel which hardens throughout in oil or water will be subject to tension stresses in the outer portion and they may actually cause rupture through quenching cracks.

While the designer is not interested greatly in those parts which actually crack in quenching, he is much interested in the intensity and distribution of stresses throughout any given part, for no formulas used by the designer cover the variation and amount of stresses incidental to heat treatment. Since, as Almen has stated, fatigue failures occur only in tension, the presence of unknown tension stresses constitute a direct hazard and usually every attempt should be made to obtain initial compression stresses if possible. It is necessary to use care in some cases to avoid excessive compression stresses in the surface zone since they must be offset by tension stresses at a greater distance from the surface. In the case of rolling contact, such as in gears and roller bearings, the fatigue failure actually may take place by subsurface tension if the surface compression loads are excessive.

Internal stresses, incidental to uneven cooling, and nonuniform transformation are aggravated greatly by nonuniform sections and by sudden changes in section. If possible these should be discussed by the designer and metallurgist before a given design is adopted. Internal

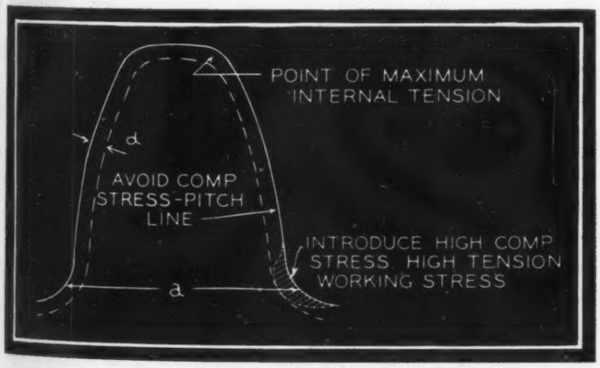


Fig. 4—In metallurgist's view of gear tooth, a should be maximum, case depth d and core hardness are governed by type of failure, and grinding, which introduces high initial tension on surface should be avoided if possible

stresses may not necessarily be severe enough to cause direct fatigue failure, but they always are the primary factor in distortion which often governs the stock removed in grinding or the performance of the given part.

Allowances usually are made for distortion in designing heat-treated parts by increased size or extra processing. This is especially true in gear tooth design when allowance is made for tooth "movement" in heat treating by extra length of tooth face and heavier tooth.

Distortion in heat treating is becoming of increasing importance to the designer because of the unknown it introduces in determining contact areas and because it indicates unknown internal stresses which make accurate designing impossible. The development of heat treatments which minimize internal stresses and resulting distortion promises to become an important factor in promoting more accurate designing.

An important contribution to our heat-treating practice was made by Shepherd[3], who determined that molten salts of certain types would cool at about the same rate as a good quenching oil. This would permit their use in cooling oil-hardening steels fast enough to insure their getting past the knee on the "S" curve untransformed. If the bath was at a temperature of 400-500 degrees Fahr. the part could be brought to this temperature uniformly throughout before transformation started. After a uniform temperature had been obtained, the part could then be cooled in air and would harden with the maximum of uniformity and the minimum of internal stresses. This practice is known as martempering.

The underlying factors governing the results obtained by the process are:

(a) Uniformity in temperature throughout a part is approached before transformation starts
(b) Transformation takes place at close to the same time throughout a given part, thereby preventing high stresses due to non-uniform volume changes.

The development of such a practice, while requiring perhaps the use of a deeper hardening steel, will permit

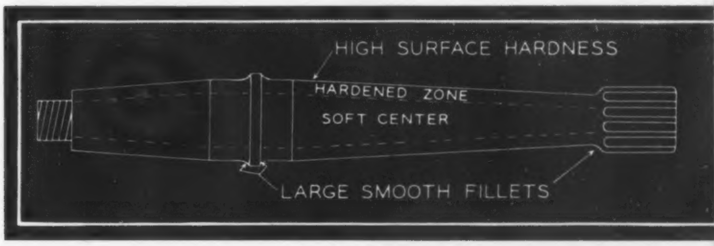


Fig. 5—As viewed by the metallurgist an axleshaft should have initial compression stresses at the surface and the surface should be free of even the most minute ruptures

the designer to assume for many parts that grinding of highly loaded surfaces can be done before heat treating, or eliminated altogether. He can assume that the uncertain internal stresses of oil or water-quenched parts can be reduced to a point where he can estimate accurately the useful strength of a given stressed area. Thus, an important step in precision designing can be taken.

Martempering in all its phases and possibilities should be developed thoroughly by the metallurgical engineer, and by him its advantages should be demonstrated and made familiar to the design engineer. There is a possibility that with martempering much of the allowance now made for distortion in heat treatment can be eliminated, and in many cases the corrective operation of grinding also can be eliminated. This will permit the designer to design more closely to the theoretical by decreasing dimensions or to increase loading on designs already in operation. Combinations of martempering, induction heating, or induction softening possibly will per-

[3] B. F. Shepherd, *Iron Age*, Jan. 28, 1943; Feb. 4, 1943.

mit designs now considered impractical. Built up combinations by welding or brazing can be made with grinding eliminated from surfaces of high hardness if this is desirable. Typical cooling-rate curves prepared by Shepherd are shown in *Fig. 3*.

The isothermal transformation of highly alloyed steels as a means of annealing is being developed at this time, and promises to reduce the aversion of production departments to this type of steel. There always has been difficulty with the more highly alloyed steels due to reduced machinability, and their use has been restricted because of it. Isothermal annealing promises to encourage their use because of the great reduction in time and the greater uniformity of the results[4].

This means that these steels might be used in many intricate parts where they offer better performance with possible reduction in size or increase in loading. If the isothermal annealing of the higher alloyed steels is combined with their better ability to process through the martempering cycle then the designer should be able to design for higher loadings than ever thought safely possible.

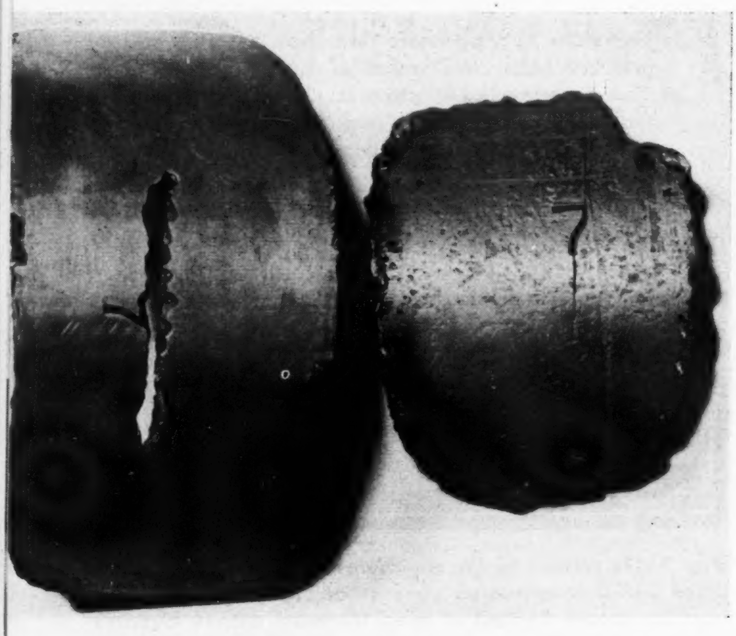


Fig. 6—Fatigue failure caused by a small surface rupture; in this case the indentation of an inspector's stamp

The martempering cycle is especially suitable to the higher alloyed steels and can be combined with short-cycle tempering baths to give maximum properties, minimum distortion and internal stresses together with the minimum overall heat-treating cycle. These developments promise to affect the design of almost every mechanism in the future. Higher strength, better contacts, and lower unit contact pressures permit smaller parts with closer spacing of supports, smaller housing sizes, and hence decreased deflection under loading.

Criterion of performance of the newer designs with increase of loading promises to be the bearings, and hence indicates possibly heavier demands on the bearing manufacturer for increased performance.

It is usual for the designer to predicate his design on his experience with parts for similar applications and where heavy fatigue stresses or severe loading are present the size of parts usually is considerably in excess of theoretical requirements. Some blame for this condition lies in the prevalence of a soft low-strength surface on most parts that are operating with applied stresses in the as-forged or as-rolled surface. This often is the case in springs, steering arms, connecting rods, bolts, etc. With ordinary practice these parts have in the unmachined surface a zone of almost carbon-free steel, with a relatively low tensile or fatigue strength. This soft surface is due to the removal of carbon by the products of combustion during heating for the various processing operations.

Combatting Decarburized Surfaces

Work done by numerous workers shows that in many applications the low strength decarburized surface is the controlling factor in the performance of the part. It is for this reason that the development of controlled-atmosphere furnaces has accelerated remarkably during the past few years. In many cases the decrease in size, resulting from machining off the decarburized surface zone, would be accompanied by an improvement in performance which possibly would pay for the machining operation.

It is believed that the studies already made indicate the potential value of the removal of the soft surface layer as a most important means of permitting a lower factor of safety and obtaining increased performance. This phase of metallurgical activity should be considered carefully by every designer of parts now giving less than the desired performance.

Production of drop forgings free from decarburization has not received the attention it should, primarily because the design engineer has not been made to appreciate its possibilities by the metallurgist.

The steel mill and drop forge furnace offers great room for improvement in respect to decarburization, but in many cases the machining away of the decarburized surface in the final part is cheaper than installing atmosphere control on the high-temperature heating furnaces in the steel mill and forge shop.

Recent developments indicate that the recarburization of low carbon or carbon-free surface zones is quite possible, and seems to offer an interesting field for improvement in performance of many applications. When subject to severe reversals of stress in bending or torsion, the strength of the surface zone is the strength of the piece, and few of us realize how low this strength is or how much money has been wasted on alloys in heat-treated parts where the criterion of performance lies in the low strength of the decarburized outer layer of the part.

As noted before, the design engineer is working continually with what is known as "strength of materials", and is primarily interested in the usable or net strength of any material under load. In the case of steel he has been taught to place great reliance on the results of laboratory tensile tests which presume to indicate the strength of the steel part in service. These tests are hardly more than academic in their value, especially in heat-treated parts. They do not indicate the internal stresses in a surface before the working load is applied. Those internal or initial stresses which are due to cold working,

[4] Peter Payson, Page 12, *Iron Age*, June and July, 1943; Payson and Klein, Page 12, *Transactions*, A S.M., 1943.

volume changes in heat treating, machining, straightening, and assembly stresses, often are far greater than the applied or working load. The useful net working strength is the difference between the internal or initial stress and the strength of the steel as indicated by the tensile test.

In some cases the available or useful strength may be considerably greater than the strength as indicated by the tensile test; in most cases it is less. To have the initial strength in a given part useful, and thereby increase the strength above that indicated by the tensile test, should be the goal of the design engineer and metallurgical engineer. This is especially true in parts subjected to severe and repeatedly applied loads where early fatigue failure threatens.

It is to accomplish this that engineers are striving to introduce initial compression stresses where highly concentrated tension stresses are developed in service. As so well explained by Almen, this can be done mechanically by cold working the surface or metallurgically by casehardening, nitriding, or the use of shallow-hardening steels.

The use of shallow-hardening steels wherever possible is a contribution to design. Their primary advantage lies in reduction of unfavorable internal stresses, and in many cases the development of favorable stresses, especially in members subjected to severe tension or bending loads which tend to cause early fatigue failures because of high tension stresses concentrated at the surface. Such parts would be axleshafts, springs, cranks, connecting rods or steering knuckles.

However, the use of shallow hardening to reduce tension stresses in the surface section should not be confused with poor or retarded quenching. In fact, to obtain the advantage of shallow hardening it is especially necessary that the structures in the hardened portion be those which indicate thorough and complete quenching. This is the first requirement for high useful strength in service. It might be said that whether he knows it or not, everyone, from the designing engineer to the final user of a critically loaded part, is more interested in the structure obtained in quenching during heat treatment than in any other operation. The war has been an important factor in concentrating attention on the part played by quenching in the final performance. Better quenching oils have been developed and more water quenching, using well worked out fixtures, has been used. Great emphasis has been placed on circulation of the quenching bath and the selection and cooling of quenching oils.

Design Benefits of Improved Quenching

As we begin to apply the experience which the war has given us in the development of the maximum properties which come with better quenched structures, the designer can begin to design closer and closer to the theoretical minimum. The maximum improvement in this direction can come only when, as pointed out by Eddy[5], the hardenability of the steel being quenched is of maximum uniformity.

To take full advantage of the improvements in control which have been developed in heat treating, and the new knowledge which is available of hardenability and time-temperature transformation characteristics, it is necessary for the design engineer, the metallurgical engineer and the operating department to cooperate in removing some factors of uncertainty which are at present all too common.

Apparently there still is some difference of opinion between metallurgists and also between designers on the methods used for arriving at the evaluation of some of these simple factors. Thus, in casehardened gears the question of case depth versus core hardness often is a matter of opinion and tradition. In *Fig.* 4 are shown some of the factors which metallurgical engineers might consider of primary importance in the initial stages of designing a gear. The metallurgist would rather see a gear fail by uniform pitting at the pitch line than by tension fatigue failure at the root of the tooth. This means that if the beam section of the tooth is sufficient the metallurgist can work out the resistance to pitting and wear by decreasing the compression load due to deep cases and

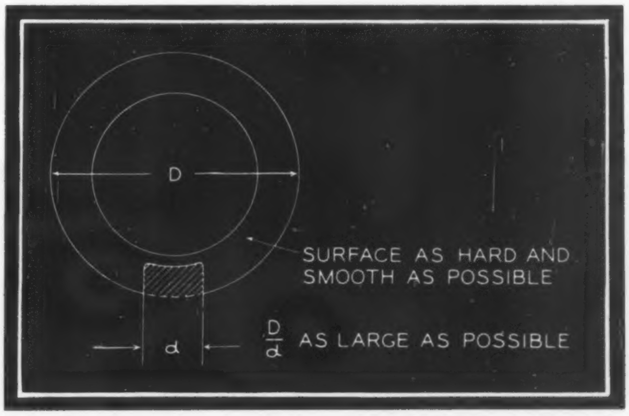


Fig. 7—Metallurgical view of worm and wheel. Wheel width d should give full center contact when fully loaded. Full-hard bronze with high yield is best for the wheel

low-strength cores. He might vary the heat treatment and core hardness, which would give him improved performance in rolling and sliding contact but which would adversely affect the strength due to high tension at the root of the tooth. This again might be due to gear tooth "movement" in heat treating or deflection in operation. The newer techniques in heat treating give promise of reducing tooth "movement". The deflection can be corrected only by design.

In *Fig.* 5 is shown an axleshaft with some notes of the factors of concern to the metallurgist. This is a torsion member which is loaded also in bending, of the semifloating type. The main concern of the metallurgist so far as performance is concerned is to start out with initial compression stresses at the surface. This means the selection of an analysis and treatment which will produce full hardness for a limited distance below the surface with a softer lower volume center. The compression stresses in the surface act to reduce the working load by whatever amount they represent. Surface of the highly stressed portion of the shaft should be free from ruptures, however minute, which would serve to start an early fatigue failure much below the estimated strength assumed by the designer. One cause of these might be cold-straightening checks; another might be an inspector's stamp, *Fig.* 6, or deep tool mark. If the shaft is not hardened throughout it is the function of the metallurgist to increase the

[5] W. P. Eddy Jr., *SAE Journal*, May, 1944.

strength in fatigue of the shaft by increasing the hardness of the fully hardened zone if more strength is desired. This usually is done by increasing the carbon content and indicates a possible trend to higher carbon steels of lower hardenability for such applications.

In *Fig.* 7 is shown a metallurgical reaction to some fundamentals of a worm and wheel. Here the newer techniques of heat treating should make possible worms of sufficient accuracy to eliminate grinding of the final hardened worm-thread surface. The important fundamental from the metallurgical standpoint is to keep the worm diameter a maximum, the worm tooth surface as smooth as possible and, most important of all, to keep the worm wheel thickness to a minimum. This restriction on wheel width is to take advantage of the improved contact at the center of the worm, the wide tolerance in dimension as compared to a wider wheel and the easier lubrication problem. Sensitivity of adjustment with worm operation seems to increase as the square of the wheel width. The problem really becomes one for the metallurgist whose function it is to recommend a material in the wheel with sufficient strength and bearing qualities

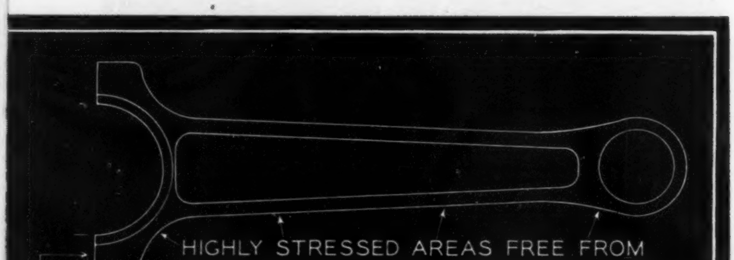


Fig. 8—Metallurgical view of connecting rod. High-tension surface is machined and shotblasted after heat treatment. Rod should be rigid in all directions

to operate with an available lubricant. This material probably is present in some of our harder high-strength heat-treated bronzes.

In *Fig.* 8 is shown the metallurgist's viewpoint in approaching a connecting rod design. Much of the same thinking is applicable to steering arms, knuckles, springs, etc., where as-forged parts are highly stressed. Here smooth machining or grinding would seem the only way to approach maximum uniformity in properties of the highly stressed surface. This is the surest way to remove low-strength decarburized surfaces and notches incidental to forging.

The effect of wartime developments in the heat treatment of steel on the production of mechanical equipment will be felt directly in the increased use of induction and flame-heating equipment which will permit line processing. The use of this type of equipment will be of value also in the relief of welding stresses in built-up parts and the localized annealing of parts to facilitate assembly.

Increased use and experience with induction and flame heating will be reflected also in its effect on casehardened applications. As induction heating progresses there probably will be a trend to use it in place of the long and in some cases more expensive casehardening operations.

There is some difference of opinion as to the relative qualities of casehardened and induction-hardened parts. The initial compression stresses introduced by casehardening are probably somewhat greater and the better wear resistance due to higher carbide concentration in the case may be a deciding factor. In any event the trend to increased application of the induction heating and flame heating process promises to continue and will affect greatly the layout of the heat-treating plant of the near future.

The use of salt bath, isothermal and martempering treatments will be of value in the processing of highly stressed parts, such as gears and axleshafts, especially in parts where distortion is a problem. This type of equipment will in most cases have to be developed as it is yet very much in the laboratory stage. Its value in reducing overall time of heat treatment, with the possible elimination of the grinding of hard, highly stressed surfaces, will in many cases outweigh its increased cost. The combining of such operations with treatments to eliminate decarburization and to control initial stresses certainly should permit designing closer to the theoretical size. This in turn will affect the whole processing picture because of perhaps drastic reduction in the size of many highly stressed parts.

New Trends in Heat Treating

As pressure is exerted to obtain the maximum useful strength in heat-treated parts, and as our knowledge of stress intensity and distribution increases, the attention of the production and metallurgical departments will be focussed more and more on continuous isothermal and salt-bath treatments for highly stressed parts. This trend also will affect markedly the planning of future heat-treating departments, and will be reflected back through every phase of steel making. Uniformity of steel will be stressed as a major requirement and may have an important effect as far back as the melting practice used in making the steel.

It is evident from the foregoing that recent developments in heat treating and metallurgical processing offer good prospects for the production of heat-treated parts with greatly decreased distortion, improved useful strength at the point of maximum loading and high hardness areas localized to insure simpler fabrication and assembly. These should, with sufficient uniformity and proper selection of steel analysis, permit decrease in size or increase in loading for many parts.

Possible improvement in load-carrying capacity can be traced directly to the improvement in our knowledge of transformation phenomena as developed in the isothermal and martempering treatments as well as to better understanding of stress intensity and location, and the control or reduction in internal or initial stresses in a heat-treated part. Improved structures resulting from better quenching, steel selection and control of decarburization also make possible better load-carrying capacity as does improved control of distortion with attendant increased areas of stress application and lower unit stresses.

Apparently we are headed for a new era in which our improved knowledge will result in an increase of the actual useful strength of highly stressed parts and thereby permit an important reduction in the "factor of ignorance".

Applying Creep Data in Design

By Joseph Marin
Pennsylvania State College

Part II—Simple Stress Members

IT WAS shown in the first article in this series how design stresses can be evaluated for specified permissible creep deformations. Before applying this information to the design of specific members it is desirable to indicate the allowable creep deformations in various constructions operating at high temperatures. TABLE II gives the allowable creep deformations that have been used in steam turbine and boiler parts, power plant equipment, steam piping and oil refining equipment. These requirements are approximate and have been selected mainly from papers published in the A.S.T.M. and A.S.M.E. *Symposium on Effect of Temperature on the Properties of Metals*[16]. To simplify the design of simple stress members, graphs giving the working stress value for particular values of the permissible creep are plotted in the following.

Axial Tension, Constant Load

In view of the inaccuracy involved in the extrapolation of creep test data and the fact that considerable test data can be approximated by the log-log method of interpretation, this method will be used in this discussion. It is true that the hyperbolic sine method may be more accurate for some test data and that it (or some other method) might well be the one eventually adopted. For the present, however, and until more test evidence is available, the log-log method appears to be sufficiently accurate.

TABLE II

Permissible Creep Rates

Application	Permissible Creep (in. per in. in 10 years)
Power Plant Equipment — Drums, tubes, headers, superheater tubes	.01
Steam Piping and Fittings	.015
Oil Refining Equipment	.10
Steam Turbine Parts	.001 to .01

16. See Pages 10, 17, 33, 55, and 146, of papers in this symposium by H. J. Kerr, F. W. Martin, E. S. Dixon, J. C. Woodson and R. W. Bailey.

For tension members subject to constant axial loads Equation 3 can be used for selecting the design stress $S = S_w$. Rewriting Equation 3,

$$S_w = \frac{1}{B^{1/n}} C^{1/n} \qquad (17)$$

For a specified allowable creep rate and for a specific material with known material constants *B* and *n*, Equation

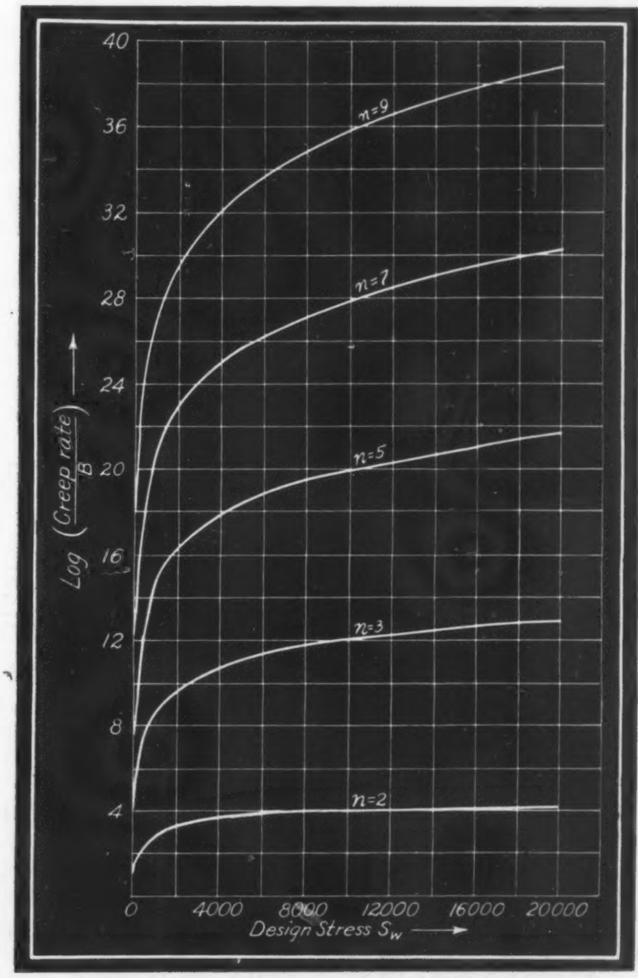


Fig. 7—Chart for selecting design stress when experimental constants and allowable creep rate are known

17 determines the design stress S_w. *Fig.* 7 is a graph representing plots of Equation 17 for range of n values usually encountered. By *Fig.* 7 the design stress can be selected when B and the allowable creep rate are known. Since $S_w A = P$, where A = the required cross-sectional area and P = the axial load, then by Equation 17 the area required is

$$A = (B/C)^{1/n} P \qquad (18)$$

Members Subject to Bending

The stresses and deflections in members subject to bending and creep have been determined both by theory and by experiment[17, 18]. Using Equation 3 for the rela-

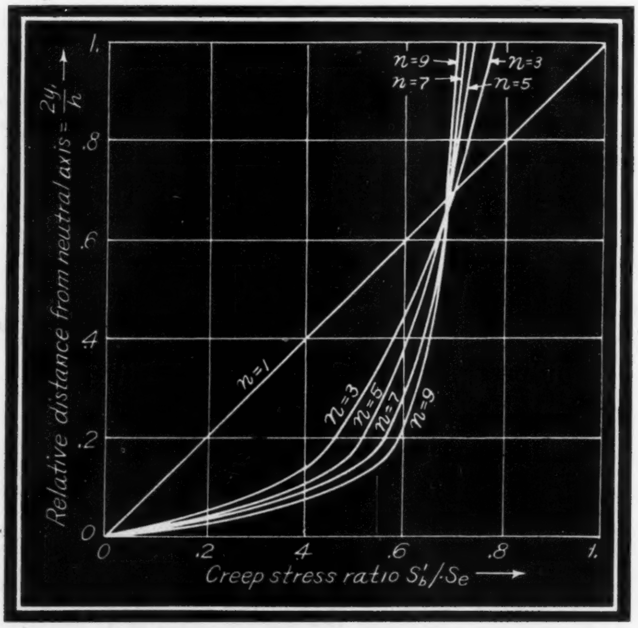


Fig. 8—Creep bending-stress distribution

tion between deformation and stress in place of Hooke's law, and following otherwise the development used in the elastic theory, expressions for the determination of stresses and deflections for constant bending loads accompanied by creep can be obtained[18]. The few test results available on creep in bending show that this theory is a good approximation. For a beam of rectangular cross section of depth h and moment of inertia I subjected to a bending moment M accompanied by creep, the bending stress S_b' at a distance y_1 from the neutral axis is:

$$S_b' = \left(\frac{My_1}{I}\right)\left(\frac{2y_1}{h}\right)^{(1-n)/n}\left(\frac{2n+1}{3n}\right) \qquad (19)$$

where n is the experimental constant of Equation 3.

The maximum bending stress is at the outer fiber where $y_1 = h/2$. Then by Equation 19, and designating S_b' at the outer fiber by S_b,

$$S_b = \left(\frac{Mh}{2I}\right)\left(\frac{2n+1}{3n}\right) \qquad (20)$$

17. McCullough, G. H.—"An Experimental and Analytical Investigation of Creep in Bending", *Transactions A.S.M.E.*, Vol. 55, Page 55, 1935.
18. For further test results and references see J. Marin and L. Zwissler—"Creep of Aluminum Subjected to Bending at Normal Temperatures", *Proceedings A.S.T.M.*, 1940.

or substituting S_e for $(Mh/2I)$,

$$S_b = S_e\left(\frac{2n+1}{3n}\right) \qquad (21)$$

where S_e = the elastic maximum bending stress.

A comparison of the elastic and creep stress distributions can be obtained by placing S_e in Equation 19 for its value $(Mh/2I)$. Then Equation 19 becomes

$$(S_b'/S_e) = \left(\frac{2y_1}{h}\right)^{1/n}\left(\frac{2n+1}{3n}\right) \qquad (22)$$

Equation 22 is plotted in *Fig.* 8 for various values of n. An examination of *Fig.* 8 shows that the maximum bending stress in the case of creep is always less than the elastic case and that the stress distribution under creep conditions is more favorable. Furthermore, a variation in the value of n has little effect upon the stress values. The important conclusion obtained from *Fig.* 8 is that if bending stresses are calculated based on the usual elastic theory $(S = Mh/2I)$ the values obtained will be greater than the actual values when creep occurs, and it is reasonable, therefore, to calculate creep bending stresses using the elastic theory. The creep deflection, however, may be a controlling factor in the design of a member subjected to bending. By using all the assumptions and conditions considered in determining elastic deflections, except the creep-stress relation of Equation 3 which is used in place of Hooke's law, a theory for creep deflections in bending can be developed[18]. For a straight member of

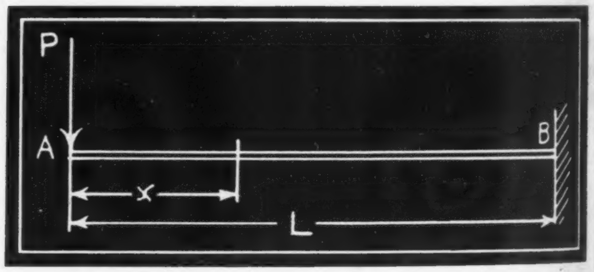


Fig. 9—Cantilever beam having an end load accompanied by creep

rectangular cross section (b by h) subjected to a moment M, the differential equation for the creep deflection y at a distance x after a time t is

$$\frac{D}{t}\,\frac{d^2y}{dx^2} = M^n \qquad (23)$$

where for a rectangular cross section

$$D = \frac{1}{B}\,\frac{(2b)^n (h/2)^{2n+1}}{(2+1/n)^n} \qquad (24)$$

Equation 23 is similar to the elastic equation $EI(d^2y/dx^2) = M$ where EI corresponds to D and $n=1$. The solution of Equation 23 for the creep deflection y depends upon the beam loading and boundary conditions.

EXAMPLE: The cantilever beam AB in *Fig.* 9 is subjected to an end load P and is accompanied by creep. To determine the end creep deflection, Equation 23 can be used if the value of moment M is expressed in terms of x. Since $M = Px$, Equation 23 can be written

$$\frac{D}{t}\frac{d^2y}{dx^2}=P^n x^n \qquad (a)$$

Integrating Equation *a* twice,

$$\frac{D}{t}\frac{dy}{dx}=\frac{P^n}{n+1}x^{n+1}+c_1 \qquad (b)$$

$$\frac{D}{t}y=\frac{p^n x^{n+2}}{(n+1)(n+2)}c_1 x+c_2 \qquad (c)$$

Using the boundary conditions of zero slope and deflection at the support *B*,

$$c_1=-\frac{P^n L^{n+1}}{(n+1)} \text{ and } c_2=\frac{P^n L^{n+2}}{(n+2)}$$

Using these values of c_1 and c_2 the creep deflection y can now be determined completely by Equation *c*. That is,

$$\frac{D}{t}y=\frac{P^n x^{n+2}}{(n+1)(n+2)}-\frac{P^n L^{n+1}x}{(n+1)}\frac{P^n L^{n+2}}{(n+2)} \qquad (d)$$

The maximum creep deflection y is the deflection desired. By inspection this deflection occurs at point *A* where $x=0$. That is,

$$y_{max}=(t/D)\frac{P^n L^{n+2}}{(n+2)} \qquad (25)$$

To determine the order of magnitude of the creep deflection compared to the elastic deflection, consider a beam with $b=1''$, $h=4''$, $L=36''$. The beam is made of a .3% carbon steel with $B=40\times10^{-38}$ (in^2/lb)^n per day and $n=7$ for a temperature of 400 C. At the end of ten years the creep deflection for an end load of 2000 pounds is by Equation 25,

$$y_{max}=.103 \text{ inch}$$

The initial elastic deflection is $y_e=PL^3/3EI$. Using a value of $E=30\times10^6$, $y_e=.195$-inch. That is, the creep deflection at the end of ten years is about 50 per cent of the elastic deflection. At the end of twenty years it would be about equal to the elastic deflection. As to whether this amount of deflection is of practical significance depends upon the particular application and the clearance to be maintained.

Members Subject to Torsion

For shafts and other members of circular cross section subjected to torsion accompanied by creep it is sometimes necessary to know the stresses and the creep angle of twist. A theory for determining these quantities can be developed using a creep rate-shear stress relation similar to Equation 3[19]. That is, a log-log relation will be assumed between the shear strain and stress, namely,

$$C_s=B_s S_s^n \qquad (26)$$

where C_s = the shear creep strain rate, S_s = the shear stress, B_s and n = the experimental constants.

19. Marin, J.—*Mechanical Properties of Materials and Design*, McGraw-Hill Book Co., 1942, Page 233.

Following the procedure and using the assumptions (except Equation 26) made in developing the elastic theory it can be shown[19] that the shear stress S_s at a radius r from the center of a bar of circular cross section is

$$S_s'=\left(\frac{Tr_1}{J}\right)\left(\frac{r}{r_1}\right)^{1/n}\left(\frac{3n+1}{4n}\right) \qquad (27)$$

where r_1 = the radius of the circular shaft, $J=\pi r_1^4/2$ = the polar moment of inertia of the cross section, and T =

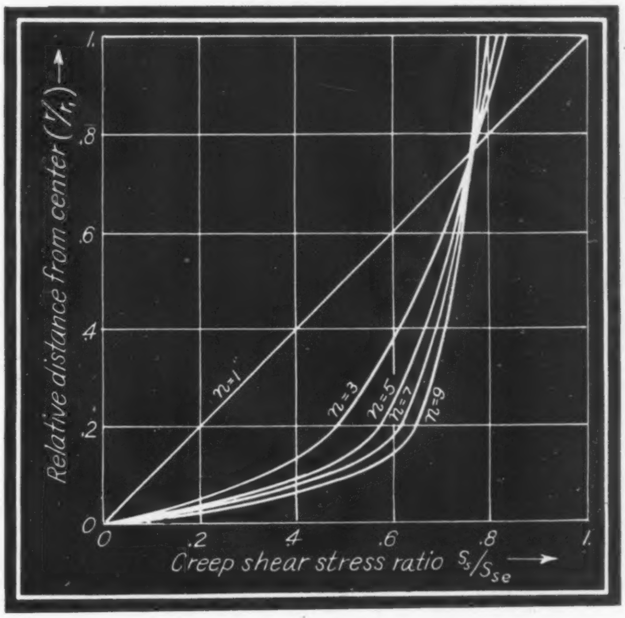


Fig. 10—Creep stress distribution in torsion

the applied twisting moment. The maximum shear stress S_s is at $r=r_1$, or by Equation 27,

$$S_s=\left(\frac{Tr_1}{J}\right)\left(\frac{3n+1}{4n}\right) \qquad (28)$$

or

$$S_s=S_{se}\left(\frac{3n+1}{4n}\right) \qquad (29)$$

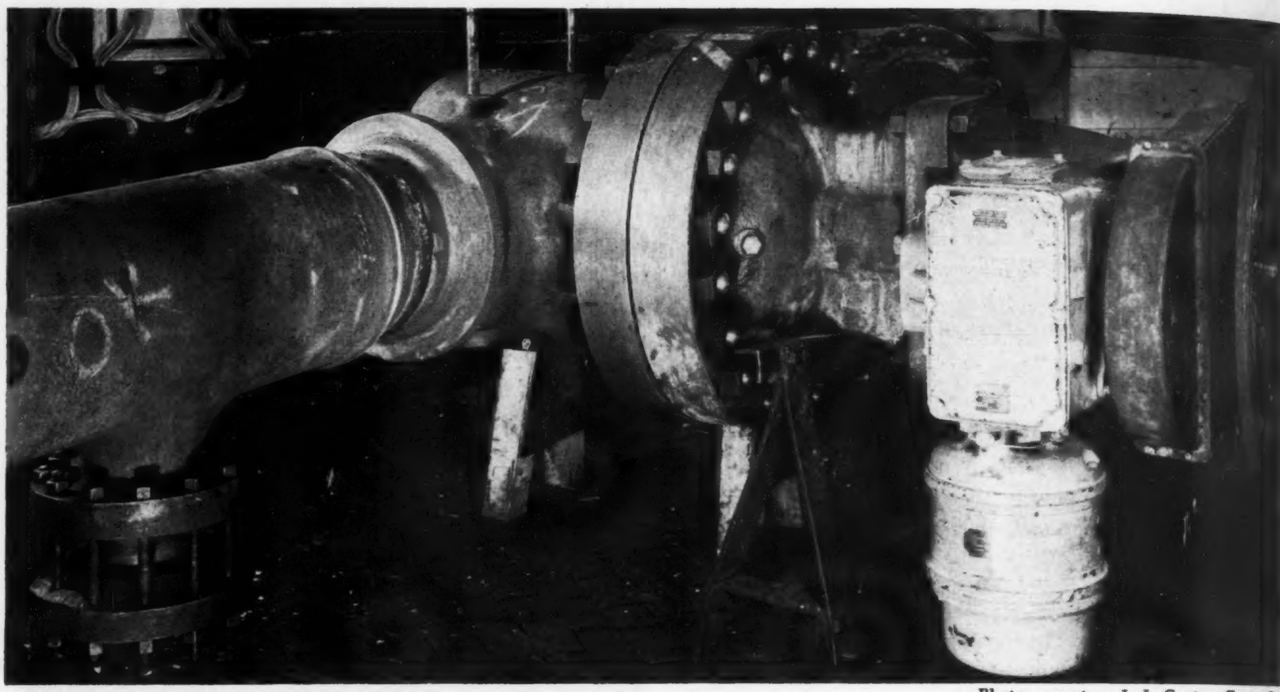
where S_{se} = the maximum elastic shear stress as usually determined.

A comparison of the elastic and creep shear stress distributions can be obtained by placing S_{se} in Equation 27 for its value (Tr_1/J). Then Equation 27 becomes

$$S_s'/S_{se}=(r/r_1)^{1/n}\left(\frac{3n+1}{4n}\right) \qquad (30)$$

Equation 30 is plotted in *Fig.* 10 for various values of n. *Fig.* 10 shows that, as in the case of the distribution of stresses in bending, the maximum creep shear stress is less than in the elastic case. That is, the creep shear stress distribution is more favorable than the elastic distribution. It appears therefore to be satisfactory if the shear stress is computed in the usual way using the elastic theory. The creep angle of twist, however, may be a significant factor.

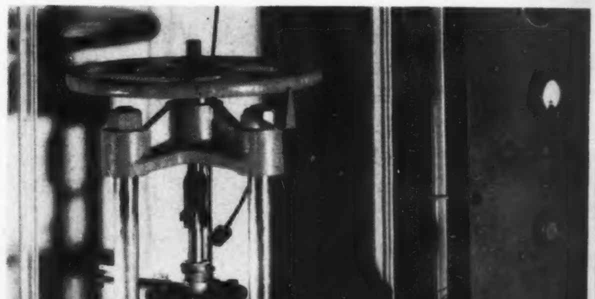
The creep angle of twist for a circular shaft subject to torsion can be determined as in the elastic case if Equation 26 is used to replace Hooke's law. Then it can be



—Photos, courtesy J. J. Cantor, Crane Co.

Fig. 11—Above—Flange and bolt connection where stress relaxation must be considered

Fig. 12—Below—Tension stress relaxation creep-testing machine in which specimen is placed in heating unit



shown[19] that the creep angle of twist per unit length of shaft is

$$\theta=\left(\frac{B_s t}{r_1}\right)\left(\frac{Tr_1}{J}\right)^n\left(\frac{3n+1}{4n}\right)^n \qquad (31)$$

By Equation 31 the creep angle of twist can be determined per unit length at the end of a time t.

Creep-stress distribution in straight members subjected to a combination of axial and bending loads has been determined by Bailey[20]. Another type of simple stress problem is that in which the stress and load do not remain constant with time. This stress condition accompanied by creep deformation is called stress relaxation.

Stress Relaxation

A number of important engineering parts creep under a condition of diminishing stress. Examples of this situation are various types of bolted joints and shrink-fit or press-fit assemblies. Sometimes stress relaxation takes place in a construction made up of several component parts.

For example, the bolts and flange combination shown in *Fig.* 11 comprises two component parts. Initially the bolts are stressed to some particular value which results in an elastic deformation in both the bolts and flange. With a high temperature and with time, a creep deformation of the flange and a tendency to deform in the bolts takes place resulting in the loosening of the flange joint and a stress reduction or stress relaxation in the bolts. If this stress reduction becomes large, serious leakage of the pipe may occur if periodic tightening of the bolts is not provided. The reduction of this maintenance is desirable and can be done by a proper selection of the initial

20. Bailey, R. W.—"The Utilization of Creep-Test Data in Engineering Design," *Institute of Mechanical Engineers (London)*, Vol. 131, 1935.

stress in the bolt. The selection of this initial stress depends upon the behavior of the material when subjected to stress relaxation. For the determination of material behavior under this loading condition many tension-stress relaxation tests have been made[21].

There are two main types of tension-creep relaxation tests, one in which the length of the specimen is maintained constant throughout the test and another in which some controlled amount of deformation is permitted. The latter type of test is similar in action to that of a bolt and flange. Tests of this kind have been made on equipment such as that shown in *Fig.* 12. In this experiment a specimen is placed in a heating unit and subjected to a particular temperature and stress. The creep deformation is measured for a specific gage length and the reduced stress is measured by determining the deformation of a weigh bar (*Fig.* 12). The data are then plotted as shown in *Fig.* 13.

Various methods have been proposed for the interpretation of stress relaxation test results. Several theories have been developed for predicting the tension-stress relaxation condition based on the constant tension load case. Using the log-log stress-creep rate relation given in Equation 3 for the constant load condition an equation for the determination of the reduced stress at a time t can be determined. It will be assumed that the length of the tension member remains constant—that is, that there is no resultant deformation. This assumption can be modified if desired.

The condition that there is no resultant deformation in the tension member may be stated by the equation

$$e = e_o' - e' \quad \text{( )}$$

where e = the tensile unit creep deformation that would occur at a time t, e_o' = the initial elastic unit deformation, and e' = the elastic unit deformation at time t. That is, Equation e states that the reduction in elastic deformation ($e_o' - e'$) equals the creep deformation. Assuming Hooke's law, the following equations apply:

$$\frac{e'}{e_o'} = \frac{S}{S_o} \text{ or } e' = (S/S_o)e_o' \quad (f)$$

where S = the unit stress at time t and S_o = the initial unit stress. Placing the value of e' from Equation f in Equation e,

$$e + e_o'\left(\frac{S}{S_o} - 1\right) = 0 \quad (g)$$

Differentiating Equation g with respect to time t,

$$\frac{de}{dt} + \left(\frac{e_o'}{S_o}\right)\left(\frac{dS}{dt}\right) = 0$$

or

$$C + \left(\frac{e_o'}{S_o}\right)\left(\frac{dS}{dt}\right) = 0 \quad (h)$$

where C = the creep rate.

Using the creep rate-stress relation of Equation 3, Equation h can be written

$$BS^n = \left(\frac{e_o'}{S_o}\right)\left(\frac{dS}{dt}\right) = 0 \quad (i)$$

Since $e_o' = S_o/E$ where E = the modulus of elasticity, Equation i becomes

$$dt = -\frac{1}{EB}\frac{dS}{S^n} \quad (j)$$

Integrating Equation j,

$$t = \left(\frac{1}{(n-1)BE}\right)\left(\frac{1}{S^{n-1}}\right)[1 - (S/S_o)^{n-1}] \quad (32)$$

Equation 32 determines the time t required for the tensile stress to decrease from an initial value S_o to a value of S. *Fig.* 14 shows the values of the stress ratio factor S/S_o for

21. For a review of such investigations see E. L. Robinson—"The Resistance to Relaxation of Materials at High Temperature," *Transactions* A.S.M.E., Vol. 60, Pages 543-554, 1938.

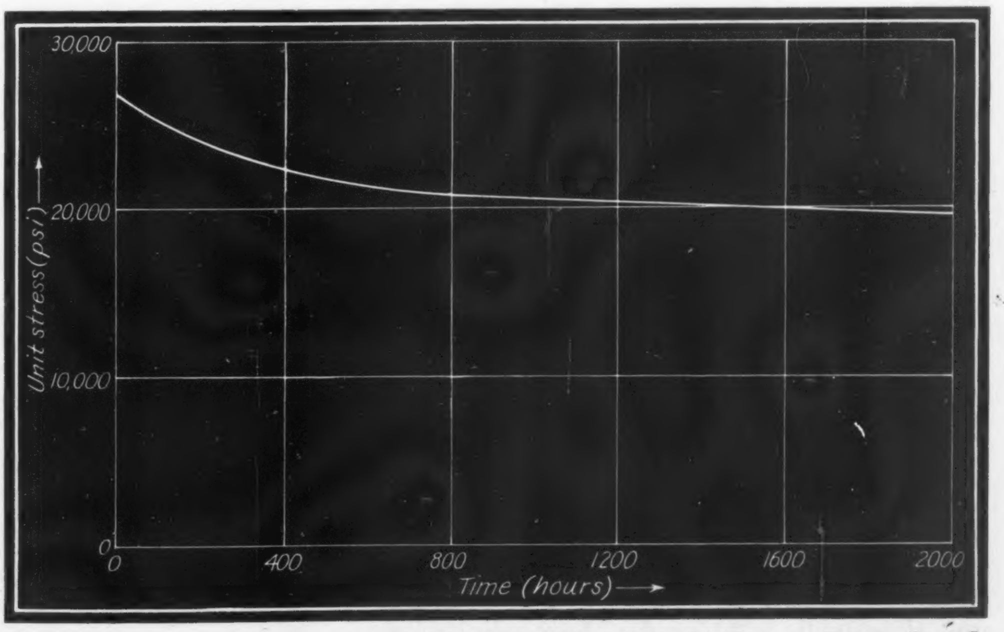


Fig. 13—Stress relaxation curve for SAE 4140 steel at 850 degrees Fahr, plotted by hours

—Tests by Crane Co.

various values of n and for an initial stress $S_o = 10{,}000$ pounds per square inch. Graphs such as *Fig.* 14 can be used for determining the possible reductions in stress.

The foregoing analysis was based on the assumption that the length of the tension member remained constant. If this is not the case, as in the bolts of the flange and bolt connection, the time required for the stress to reduce from

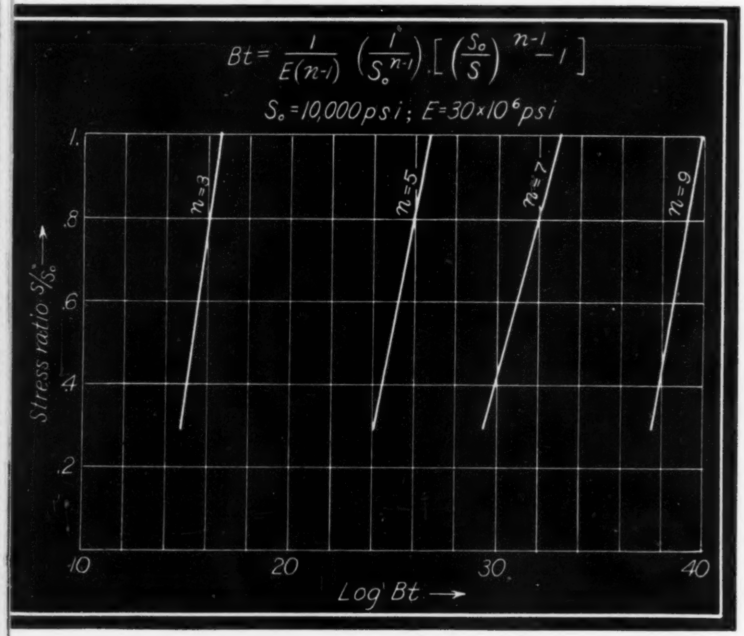


Fig. 14—Tension stress relaxation diagram

S_o to S is less than that given by Equation 32. Then the time can be found by multiplying Equation 32 by a factor K. That is,

$$t = \left(\frac{K}{(n-1)BE}\right)\left(\frac{1}{S^{n-1}}\right)[1-(S/S_o)^{n-1}] \quad \ldots\ldots\ldots (33)$$

The value of K in Equation 33 can be determined if the relative elasticity of the component parts is known.

The Artists' Dream Car

THERE has been much talk and prophecy in connection with increased vision in the postwar automobile. The corner windshield and door post long has been a bugaboo not only to the driver of the car, but also to the designer and manufacturer. Yet this condition cannot possibly be as easily solved as our prophets indicate in the one-piece, molded, teardropped, plastic roof. These artists, and I use that term for it is their greatest claim to fame, undoubtedly have been exposed to pictures of bombers utilizing plastic noses, gunners' turrets, and observation bubbles on the fuselage of the plane.

Without, apparently, any further question or checking of the cost and methods of fabrication used in these aircraft elements, the artist prophet blithely translates the generalities of the idea into the motor car, showing if you please, a one-piece molded section comprising the roof, windows and windshield all in one. No struts, or supporting ribs, or unsightly rivets, or anything as unromantic as that are used.

To go further, I have never been able to understand how we would keep the passengers from having heat prostration under this transparent top. The sun's rays certainly would penetrate beyond any of the known benefits of existing air-conditioning equipment. The plastic prophet dismisses this with a glib statement that new plastics will filter out all injurious heat rays, providing only an enviable tan.

In this virtual goldfish bowl, despite its supposed advantages for vision, I am quite sure there would be a strange change of feeling even between riding in the accepted convertible coupe with its top thrown back, as against this hermetically-sealed showcase. In these picturizations, the artists have eliminated the corner post problem and have offered panoramic vision with a ball-shape windshield to complete the aerodynamic conception. I have yet to find a proposed design of this type that has considered that drab little problem of windshield wipers. That is finally solved, as far as the illustration is concerned, by leaving them off. Perhaps they are left off the illustration because even the artist realizes that the normal swinging-type wiper of reasonable manufacturing cost will not wipe a compound curved surface. Further, the plastic material available is still not suitable for windshields because the surface hardness is not sufficient to withstand windshield wiper action in conjunction with the grit and dirt present in rain.

Engineers To Work Out Problems

The last jarring detail in connection with the transparent roof is the showing of a Lucite window pane, within the plastic door, closing with its surface flush with that of the top. Imagine one die-cut edge of Lucite meeting the other die-cut edge of the Lucite top and not leaking. I am afraid we would have to call upon the highest powers to eliminate rain before this type of thing would be practical. I might add that, generally, when the authors of these prophecies are questioned in regard to any of these points, they dismiss them with a wave of their hands and say that the engineers will work those things out. Gentlemen—are you ready?

It is entirely possible that some increase in vision could be accomplished in the early postwar models through revisions in the existing designs. If we were to widen the body further at the cowl and move the heavy door and corner post outward and back from its former position, it might then allow for more nearly 180-degree driver vision through the use of two safety-glass windshield panels divided in the center by the usual mullion—each windshield panel to have a single plane bend utilizing a section of not less than 60-inch radius at the three-quarter point to accomplish this partial encirclement. This curvature has proved to be the minimum without distortion of vision. Windshield wipers could still be operated on the flat surfaces.—*From a talk by Brooks Stevens, presented at the National War Materiel meeting of the* S.A.E. *in Detroit.*

Strength of New Plastics

Compares with That of Metal

• By John Delmonte
Technical Director
Plastics Industries Technical Institute

A NUMBER of new liquid thermosetting resins have made their appearance during the past two years, and have proved their usefulness in extending the industrial applications of plastics. The chemical origins of these resins are varied and, in some instances, new to the field of plastics. However, they possess certain similarities in physical characteristics, including clear, colorless transparency and the ability to produce laminated structures. They are distinguished further by their anhydrous nature and by their ability to be hardened by organic peroxide type catalysts such as benzoyl or lauroyl peroxide.

The laminates which have been developed from these resins are of unusual interest to machine designers because they have made possible materials with strength-weight ratios comparable to aluminum alloys, together with fabrication techniques which permit the development of complicated structures in a single operation. The new resins are being used to impregnate glass fabrics or cotton fabrics which are assembled in a suitable form and cured "en masse" with heat and low pressure. Plastics and glass fibers are a particularly unique combination. Glass fibers may be drawn to extremely high tensile strengths and woven into fabrics. With the aid of resin binders these fibers are consolidated into solid sheets still retaining the physical advantages of the high-strength fibers.

Fig. 1—Experimental plane employs balsa as a core material with face plies of glass fabric laminate

Considerable research work has been carried out in the laboratories of glass manufacturers on the development of high-strength glass fibers. Tensile strengths of fibers in excess of 2,000,000 pounds per square inch have been attained experimentally, though when woven into fabics and resin-bonded into sheets, the ultimate values are appreciably lowered. Highest ultimate strengths, however, have been realized from special fabrics which are laminated together so that there is a preferred orientation of the high-strength fibers.

Resin-bonded glass laminates have been used in conjunction with other materials to achieve special characteristics. For example, glass fabric laminates used as face plies and balsa as a core material contribute to a high

strength-weight ratio. Advantage is taken of this unusual construction in an experimental airplane, *Fig.* 1, built at Wright Field.

From the machine designer's viewpoint this example is unique in that it is an outstanding example of stressed plastic parts, whereas the working stresses of the more common molded and laminated plastics are quite low, lying under 1000 pounds per square inch for most thermoplastics and under 2500 pounds per square inch for most thermosetting plastics, when based upon elastic limit at room temperature.

Several manufacturers are contributing actively to the development of these new resins, and in order to keep readers of MACHINE DESIGN informed of the latest materials, TABLE I has been prepared to aid their identification.

APPLICATION OF RESINS TO LAMINATES: Before the resins are applied to glass or cotton cloth they must be activated through the addition of an organic peroxide as a catalyst. Best known of these are lauroyl peroxide or benzoyl peroxide. The resin is generally warmed slightly to aid the dissolution of the catalyst. After the catalyst has been added the glass or cotton fabrics are treated with the resins either by brushing the resins into place or by a continuous process involving progressive immersion of the fabric and subsequent wiping off of excess material. The resin is furnished either in a liquid or paste form.

TABLE I

New Liquid Thermosetting Resins

Tradename	Types	Where Used	Manufacturer
Columbia Resins..	CR-38, 39	Transparent housings and laminates	Pittsburgh Plate Glass Co.
Allymer Resins ...	CR-139	Laminates	Pittsburgh Plate Glass Co.
Laminac	X4000, X4122	Laminates and transparent sheets	American Cyanamid Co.
XR 16631	....	Laminates	Carbide & Carbon Chem. Corp.
Plaskon 900	....	Laminates	Plaskon Co. Inc.
Monsanto 38691 ..	....	Laminates	Monsanto Chemical Co.
MR Resins	MR-1, 1A, 17	Laminates	Marco Chemicals Inc.

The paste form is sometimes preferred because the thinner, more liquid resin tends to run off the glass or cloth during initial heating prior to final polymerization. It also is the practice to include a small percentage of flock fibers with the resin impregnant in order to increase edgewise compressive strength. Maximum physical properties are reached when the fine flock equals 3 to 4 per cent of the weight of the resin present. It also may be desirable to heat glass fabrics at 200 degrees Cent. for two hours in order to remove surface lubricants prior to impregnation. Improvements in some physical properties are realized, though impact strength is lowered.

Handling and curing of the impregnated sheets are largely dependent upon the size and shape to be developed. For example, if flat laminates are being prepared, air bubbles are excluded by rollers acting from the center outward. The sheets are then covered with cellophane or glassine paper and placed between polished metal sheets or glass plates. Function of the cellophane is not only to prevent sticking to the plates but also to keep the materials out of contact with the air. Curved shapes built up from laminates are prepared by cutting the fabric in accordance with templates and then laying the pieces up on wooden, plaster, or cast metal forms. One of the attractive features of these operations from a designer's viewpoint is the absence of costly molds and presses which are necessary for the more conventional molding or laminating operations upon phenolic or urea plastics. Typical laminated parts made with these high-strength resins are shown in *Figs.* 2, 3, 4 and 5.

Attractiveness of these new liquid thermosetting resins lies in the fact that they are intended for very low-pressure laminating. In fact, various parts and equipment now are being prepared in which no more pressure is applied than that obtained under vacuum, opening up the possibility of using wood or concrete molds. The building up of complicated machine bodies from laminated plastics appears less remote with the advent of this new technique.

Fig. 2—Employing a liquid thermosetting resin, this laminated part is a fillet for an army training plane

CURING CONDITIONS: The resins discussed in this article are best cured out of contact with air, otherwise the materials may not fully cure. Temperatures of 240 to 300 degrees Fahr. are applied generally for a period of time depending upon the thickness, though cure times of 30 to 60 minutes are not uncommon. The resulting product converts over to an infusible, insoluble thermosetting body while curing and acquires some of the excellent physical properties listed in TABLE II. When properly made there is no evidence of any air or gas entrapments, and no tendency to delaminate if an effort is made to pull the layers apart. Resin contents vary from as low as 40 per cent for some glass laminates up to 65 per cent for some cloth laminates.

DISADVANTAGES: Critical examination of the new low-

pressure laminating techniques as made possible by new liquid thermosetting resins reveals difficulties which may be objectionable under certain conditions. For example, operators handling some of the liquid resins may be allergic to the chemicals present, although this minor factor has not discouraged activity.

There is a tendency among fabricators of these laminated parts to use no more pressure than absolutely necessary—contact pressure, if possible. As the laminating pressures go down a tendency increases for air or gas pockets to develop. Likewise the tendency to

TABLE II

Properties of New Glass Laminates Compared with Other Materials

Property	Glass Laminates* New Liquid Thermosetting Resins	Grade C** Laminated Phenolic	Aluminum Alloy† 24ST
Specific Gravity	1.70 to 1.77	1.34	2.7
Ultimate Tensile Str., psi	45,000 to 55,000	9,500	68,000
Compressive Str., psi	40,000 to 55,000	38,000	.....
Flexural Str., psi	60,000 to 85,000	20,000	20,000
Modulus of Elasticity in Flexure, psi	$2.0 \times 2.5 \times 10^6$	1.0×10^6	10.0×10^6
Proportional Limit, psi	23,000 to 30,000	2,500	32,000
Water Absorption, %	.3 to .5	.7 to 3.0	0

*These are average of several resins in combination with glass cloth.
**These are typical values largely reported by Plastics Materials Mfrs. Ass'n.
†From "Mechanical Properties of Metal Alloys"—C-447, U. S. Dept. of Commerce, Bureau of Standards

Fig. 3—Plane parts, such as this engine fairing, are being laminated with new high-strength resins

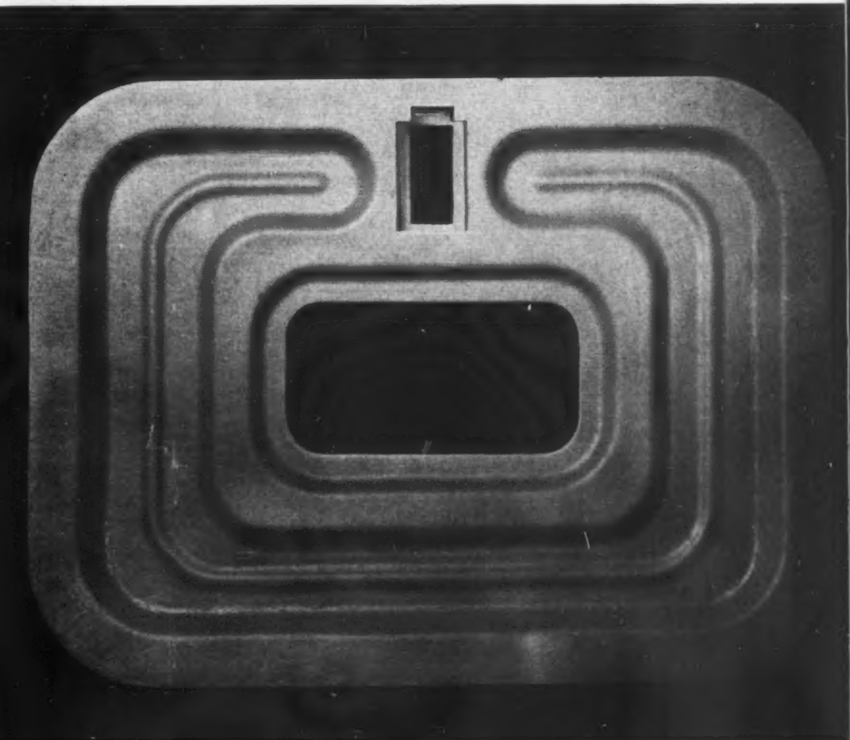


Fig. 4—This is the side gunner's door for an army bomber, molded with one of the new resins

delaminate may be much more pronounced. There are other fabricating problems more significant than these. High-strength glass fiber laminates depend upon perfect orientation of the glass fibers for optimum results. At best this is achieved only in fibers which have been carefully combed and laid parallel just prior to laminating.

While some of the new resins, such as CR-149, have made a remarkable gain in physical properties, without any reinforcement of fibers, due to the careful chemical research which pointed the way to improved strength in organic structures, the importance of the glass fabrics in converting these resins to structural laminates should not be minimized. One cannot overlook, however, the cost of glass fabrics—over $1.50 per yard—and the expense of the subsequent laminate.

PROPERTIES: Some of the outstanding physical properties of glass fabric laminates which will interest machine designers are listed in TABLE II. A metal is offered for comparison, together with a more conventional Grade C canvas-base phenolic laminate. The values for the glass laminates were taken longitudinally to the main fiber direction.

TRANSPARENT VARIETIES: Another field of application of some of the new oxygen-hardening thermosetting resins which will interest machine designers is the preparation of transparent enclosures capable of withstanding higher operating temperatures than the previously employed transparent thermoplastic materials. Prior to the war, for example, there were no transparent organic plastics quite capable of withstanding the temperature of boiling water. Cast transparent sheets of some of the ester or allyl type thermosetting resins now are being used as transparent enclosures requiring heat resistance superior to the present thermoplastics. These transparent thermosetting sheets also are being produced in various colors.

APPLICATIONS: The physical data contained in TABLE II speak for themselves in indicating the tremendous strides taken by the new resins. Experimental values

notably greater are reported for glass fabric laminates.

In general, three major fields of application have developed for these resins:

1. Flat laminates—low contact pressure
2. Contoured laminates
3. Transparent cast sheets.

It is interesting to note that flat glass laminates bonded with MR resins now are being produced by a continuous

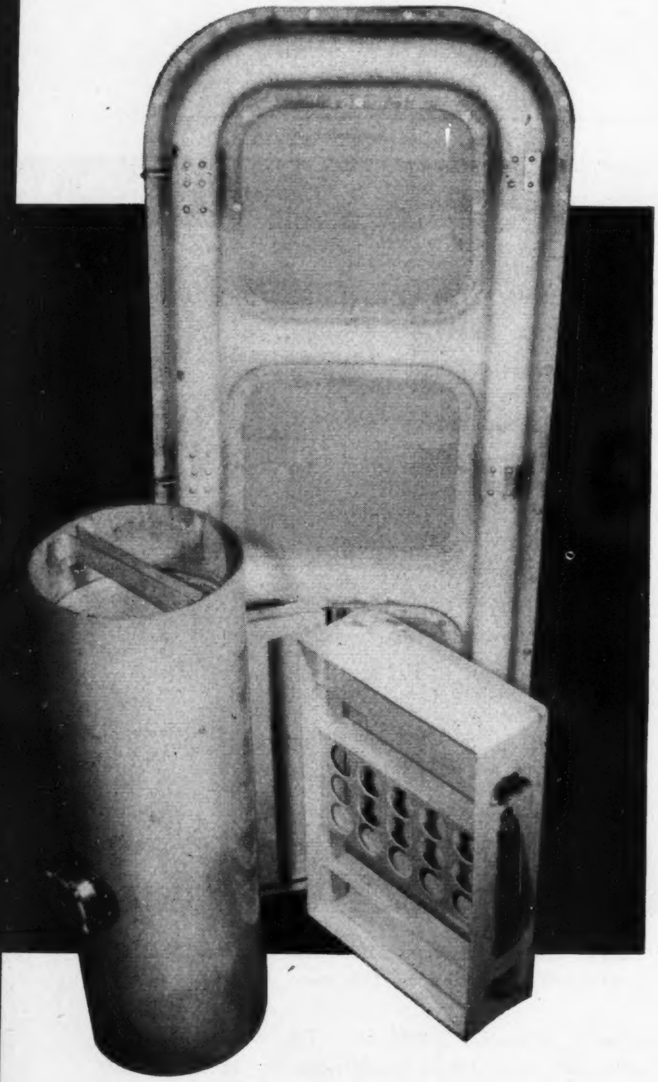


Fig. 5—Included in this picture are three parts laminated with liquid thermosetting resins—a navy water breaker, an arctic first-aid kit enclosure and a life-raft door for an army bomber. All require light weight and high strength

process. This development will expedite the production and further the applications of this material.

All of the foregoing forms of these new materials will be of interest to designers of machines and will extend the scope of their efforts. Of course competitive laminating techniques, such as shown in *Fig.* 6, must be taken into account. In the figure, the cloth-laminated container has been built up at low pressure, using thermoplastics in solution form applied by brush. The chief disadvantage is the time lost in waiting for solvent evaporation, but the resulting tough thermoplastic laminates show much promise.

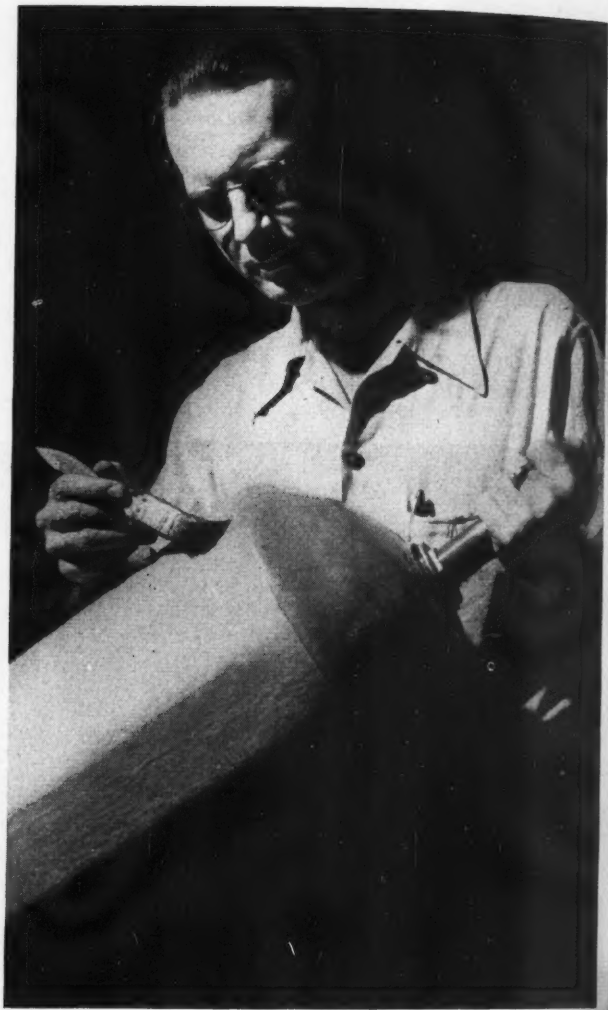


Fig. 6—Container made from cloth laminate is built up at low pressure, using thermoplastic in solution form applied by a brush

Development in the new thermosetting resins is far from complete, but it is important at this time to recognize the trend. With the aid of these materials, interest will grow rapidly in structural applications of plastics.

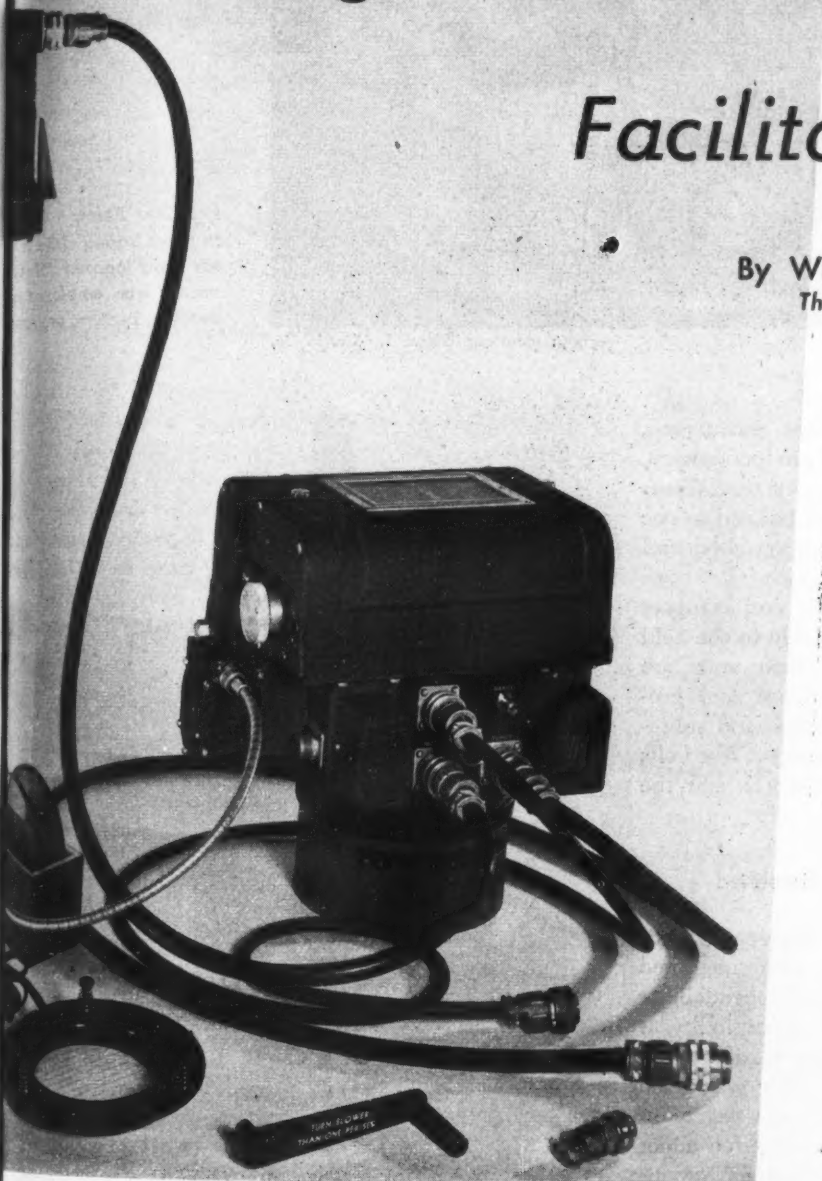
Illustrations used in this article were furnished through the courtesy of American Cyanamid Co. (*Fig.* 3); Army Air Forces, *Fig.* 1; and Marco Chemicals Inc. (*Figs.* 2, 4 and 5).

AT LEAST TWO AMERICANS invented flying bombs, similar to the German robot planes, as long ago as the last war. Mr. Lawrence B. Sperry in 1918 filed application for a patent covering a gyroscopically stabilized aerial torpedo of the airplane type, propelled by an electric motor.

In 1919 Dr. Charles F. Kettering applied for a patent on a similar type of flying bomb, propelled by a two-cycle gasoline engine and designed to carry several hundred pounds of explosives. Direction and altitude in flight were gyroscopically controlled and the explosive charge was to be detonated after a predetermined flying distance. Launching was to be effected by means of a catapult device on an inclined plane. Patents were subsequently granted on both designs.

"Package" Motor-Control Units Facilitate Design

By William H. Fromm
The Dumore Co.



—Photo, courtesy Folmer Graflex Corp.

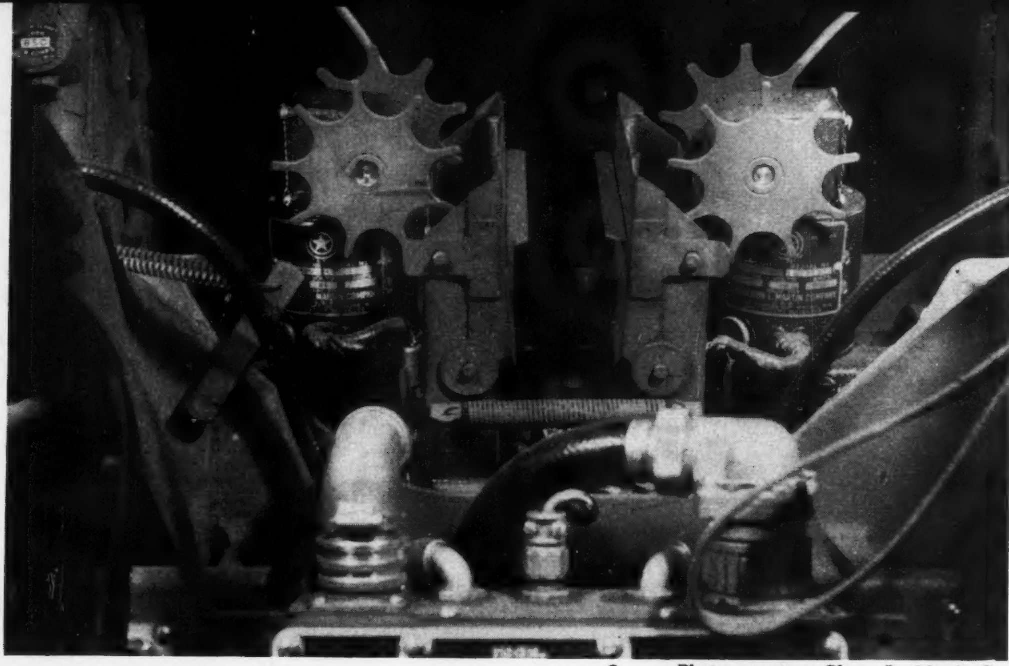
Fig. 1—Aerial camera utilizing geared motor with magnetic clutch and brake, driving through flexible shaft

Part I—Motion Control

ALTHOUGH the use of small electric motors on domestic, industrial and commercial machines and devices has been widespread, fractional horsepower motor applications will increase and broaden immeasurably when peace returns. Machines now operated manually, machines now on the drawing board, and the myriad of machines yet to be designed will draw their power from these compact and efficient motors. War applications in combat equipment have dramatically proved their dependability and versatility, and tomorrow's machines will inherit a new utility and convenience—a legacy of wartime advancements in engineering design which telescoped years of progress into a few months of achievement.

Naturally, many motor-drive applications require no control or auxiliary equipment. In this category, for example, are blowers, routers, drill presses, and vacuum cleaners. However, an impressive majority of machines must have intermediate or auxiliary devices and controls to modify the motor power output. Speed must be reduced, varied or otherwise controlled; coasting must be eliminated; operation must be terminated at a given limit; motor and device must be protected against possible jamming; and provision must be made for a variety of internal and external conditions which may act upon motor or machine efficiency.

In the past it was the practice in most instances to install auxiliary controls and equipment separately, rather than as an integral step in the process of manufacturing. All machine manufacturers were customarily obliged to purchase or build speed controls, overload protectors, switches and similar devices separately and assemble them, with the motor, to the machine. Disadvantages inherent in this practice are obvious: Divided responsibility; weight, space and cost penalties; and design, procurement and production difficulties. The swing toward packaged power units with all neces-



—Photo, courtesy Glenn L. Martin Co.

Fig. 2—Left—Drive fo ammunition booster i aircraft gun turret utilize motors having combina tion spur and worm-ge units with built-in ove running clutch

Fig. 3—Below—Gove nor-controlled drive m tor and geared take-u motor are employed i motion picture project

sary controls built-in was at least strongly encouraged, if not demanded, by design engineers and production men. They saw the manifold advantages of integrated drive: Responsibility for satisfactory operation centralized in one supplier; weight, size and cost reduced appreciably; and assembly simplified into a single operation.

Wartime applications demanded simple and compact power drives which could be serviced quickly in the field by relatively inexperienced personnel. Such units are complete, all-in-one power packages designed and produced to the machine designer's specifications and subsequently installed in his assembly by fastening a few bolts and nuts, and connecting to the energy source and the driven device.

Assembly and Maintenance Are Simplified

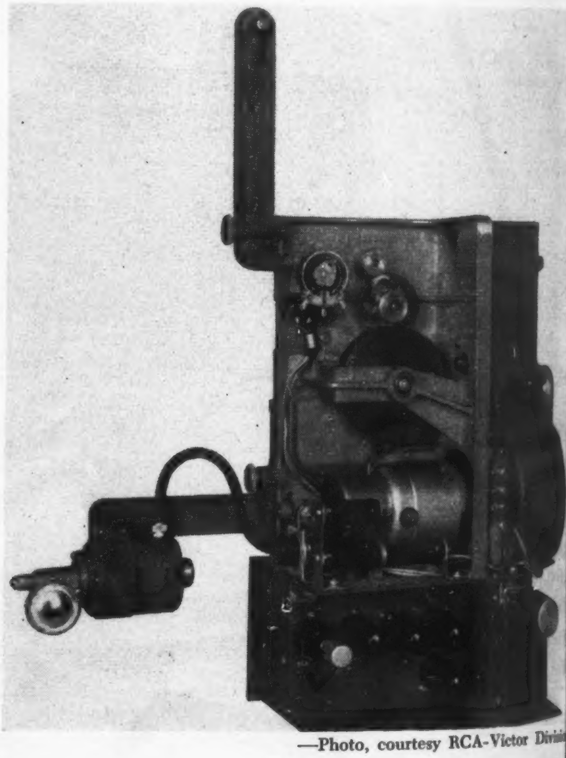
Aircraft manufacturers realized, even before the war, the desirability of simplifying assembly with integrated units, and (considering the importance of dependability to aircraft) the importance of undivided responsibility as well. Their insistence that all necessary auxiliary devices be incorporated into the motor assembly was given a new weight and urgency by the sudden necessity of servicing military airplanes all over the world, often under primitive conditions. An airplane grounded for service on a front line airfield was out of action for a period measurable by the complexity of maintenance instructions and number of spare parts that servicing involved.

Today many complete power drives are little larger than prewar motors by themselves. Relatively complex in their construction, the new drives simplify tremendously the actuation of such devices as cowl flaps, oil-cooler doors, retractable landing gear, aerial cameras (*Fig.* 1) and ammunition boosters (*Fig.* 2).

Motor auxiliaries may, for consideration, be grouped according to four distinct functions:

1. Motion control
2. Motion modification
3. Protection
4. Indication.

Continuous development on integral auxiliaries may add



—Photo, courtesy RCA-Victor Divi

new categories but at present the four are inclusive.

Rheostats: Rheostats have been used to control mo tor speed for years. Although they have not generall been supplied as an integral part of the motor, applica tions requiring frequent speed variation might well mak use of such a combination.

Essentially a coil of high-resistance wire—chrome manganin, nichrome, etc.—with a means of shorting o all or part of the coil, a rheostat provides speed variatio by varying the voltage applied to the motor or to part o the motor windings. The applied voltage is reduced b an amount equal to the product of the resistance of th rheostat coil in ohms and the current in amperes flowin in the motor circuit. Because the power loss (which i the product of the current and the voltage drop acro the rheostat) appears as heat, the wire is frequen

wound on a heat-resistant core, usually of some vitreous or ceramic material. The necessity of dissipating this heat is a factor to be considered when a rheostat is designed as a built-in motor component.

This type of motion control is best applied to either shunt or series motors but can also be used on shaded-pole motors. Rheostats may be connected with motors in a variety of methods and combinations, involving either field or armature or both. The method used is determined by the amount of speed variation required, the permissible no-load speed, the degree of speed stability required and the amount of power loss acceptable. Rheostat control of any type, however, is somewhat inefficient.

Motor applications which could make use of built-in rheostat control include motion picture projectors, winding machines, fans, blowers, centrifuges, and dental lathes.

Electrical Governors Give Good Control

GOVERNORS: The limiting of small-power motor speeds with governors is by no means new, but new possibilities in their application have become apparent in recent development and experimentation.

Two principal types of governors are mechanical and electrical. The usual type of mechanical governor has spring-mounted, centrifugally-actuated weights which apply a friction brake when the speed of the controlled motor reaches a predetermined value. Although the action is nonfluctuating the governor is wasteful because speed limitation is accomplished through friction. Consequently this type is used only on relatively low-power and

The electric governor appears in several variations, of which the most familiar one is the centrifugal type with spring-mounted contacts connected in the motor circuit. When motor speed reaches a predetermined value, centrifugal force opens the contacts—which usually are carried on the motor shaft—and interrupts the motor circuit. As the motor begins to slow down past the predetermined high value the contacts close once more to re-establish the circuit. The result is an extremely rapid but minor fluctuation (imperceptible beyond very low speeds) near the desired speed value.

Another type of electrical governor control employs a magnet which is mounted on the motor shaft and rotates within a thin metal cup or keeper. Eddy currents set up in the keeper tend to make it rotate in the same direction as the magnet. This tendency is transmitted through suitable linkages to a set of contacts, which are in the motor circuit. When the force of the rotating keeper becomes strong enough, these contacts open and the motor slows down, subsequently permitting the contacts to close and the cycle to repeat. Here again speed fluctuation is rapid enough to be discernible only at very low speeds.

Another type of governor is that which has its contacts actuated by a blast of air from a fan or a blower wheel on the armature shaft. The air pressure set up by the blower acts on a spring-loaded diaphragm or butterfly plate which in turn causes the contacts to vibrate in the same manner as on the centrifugal governor.

In all electric governors the limiting factor is the action of the contacts which make and break line current. To reduce arcing a combination of resistors and condensers usually is shunted across the contacts, and under certain circumstances good contact life can be obtained. In general, however, electric governors have not been too satisfactory on heavy currents (particularly direct current).

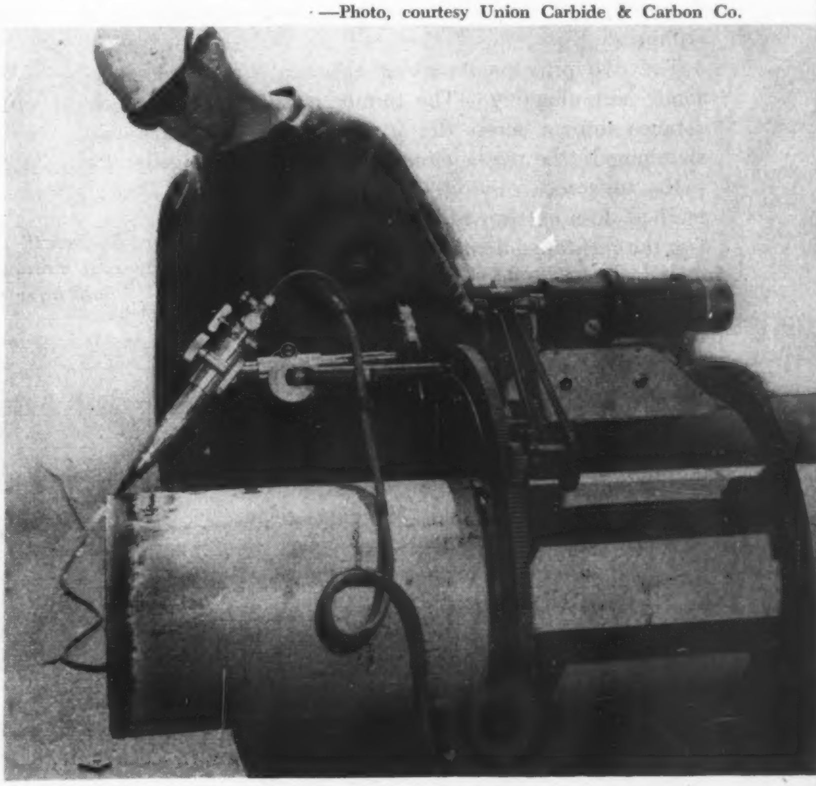
Nearly all governors, electrical and mechanical, are adjustable (many while the motor is in operation) by the turning of a screw. Speed may be varied over a range in the neighborhood of 5:1 and will remain constant within plus or minus 3 per cent or less at any setting within that range. The governor, however, is a speed-limiting device only and will not prevent underspeed should the load increase beyond the motor rating or the line voltage drop.

Governors are used principally on motors of the so-called "universal" type to provide, in a motor of small frame size, the constant-speed characteristic of the larger induction motor. Governor applications were increasing noticeably before the war and as improvement continues and new types are developed it is logical to assume they will be employed even more generally. Their principal wartime use is on photographic equipment—sound motion picture cameras, projectors (*Fig.* 3), intervalometers (a precision instrument for timing aerial photographs) and other timing equipment. Peacetime equipment which may utilize the unique characteristics of governor-controlled motors are: Portable motion picture projectors, kitchen mixers, recorders, certain types of office ma-

VARIABLE-SPEED TRANSMISSION: In addition to rheostats and governors there is a third device used to vary the speed of fractional horsepower motors: The small transmission in which motor speed is changed by means

Fig. 4—Flame-cutting machine has motor with double-reduction worm-gear unit and electric governor

—Photo, courtesy Union Carbide & Carbon Co.



of variable pulleys, tapered rollers running inside a ring, or other similar mechanisms. Large transmissions have long been used on machine tools, ovens, paper machines, etc., but a unit for fractional horsepower motors is a recent development. The newly devised transmission, *Fig.* 5, is controlled by a crank which allows speed variation over a wide range, and in addition provides mechanical reversal of the direction of rotation. One precision wartime instrument employs a governor-controlled motor driving through a variable-speed transmission to furnish a necessary combination of constant motor speed and adjustable output speed.

Although more expensive and more bulky than rheostat or governor control, the variable transmission is free of the torque loss associated with the former and the high maintenance requirement of the latter.

Brakes: Not the least important requirement of current devices and postwar designs is quick stoppage to eliminate the normal coasting of the driving motor when the circuit is broken. Precise positioning of operating elements which necessarily is a feature of many machines requires the elimination of all appreciable overrun. To absorb the kinetic energy of the rotating mass of the motor several effective quick-stoppage devices have been evolved in the form of electromagnetic, electric and mechanical brakes.

Motor Brakes Have Become Essential

In the electrification of cowl flaps or wing flaps or landing gear, the aircraft designer was confronted with the problem of preventing normal motor overrun (which might vary somewhat from motor to motor and with atmospheric conditions) from jamming or deforming the actuating members after the limit switches had opened the circuit. An automatic brake on the motor was an essential. The urgent requirement was met and today nearly every type of actuating motor on new airplanes is so equipped.

The two principal kinds of electrical braking are dynamic and plugging. The former usually consists of resistance thrown across the armature with field excitation maintained (the motor momentarily acting as a generator —the resistance serving as a load). This method does not provide rapid braking unless the resistance across the armature is reduced almost to zero, and such a reduction produces high currents detrimental to windings, commutators and brushes. A system of relays is required. Dynamic braking is most useful for gradual braking.

Plugging, the second type of electric braking, involves the momentary reversing of the polarity of the motor windings. Also available is a so-called plugging relay, which reverses the windings and opens the circuit at the precise instant the rotor speed reaches zero. Plugging for quick stoppage naturally produces high currents in the windings and if used repeatedly would cause the motor to overheat. Neither elec- method, however, will hold the motor shaft once it has stopped.

Mechanical brakes, while fairly satisfactory, are difficult to make automatic. The manually operated friction brake is the simplest type but the complexities involved in providing remote operation for their application to small motors precludes the consideration of mechanical brakes for motors in this range.

The electromagnetic or, more simply, magnetic brake, *Fig.* 6, is most suitable and practical for incorporation in fractional-horsepower motors. Basically it consists of two friction surfaces held apart by magnetic force while the motor circuit is energized, and forced together by spring action when the circuit is opened. A variation, which may be used when the motor is in operation for longer periods than it is standing idle, is to hold the surfaces apart by spring action and engage them magnetically. In either case the stopping time, which is a function of the tractive effort of the magnet, is determined by the size and design of the springs and the number of ampere turns of the magnet winding.

When the design utilizes magnetic force to hold the brake open, the magnet winding may be connected in series with the motor winding to insure (through the high starting current) immediate release upon starting. Even better, however, is a compound winding—one coil in series with the motor and one coil across the line. This arrangement provides positive brake action on starting and under load as well as under light loads when the current flowing in the series coil is low. The brake winding of course consumes some power and therefore slightly reduces the overall efficiency of the motor.

The three principal types of brakes, mechanically speaking, are disk, internal-expanding and external-contracting. The first employs a disk mounted on the armature shaft and engaging a surface fixed to the motor frame. In some designs the disk is fixed rigidly to the shaft and the entire armature moves axially, but a superior design couples the disk to the shaft through a spline. This coupling permits a positive radial connection and leaves the disk free to move axially while the armature retains its normal position. Still another type of design provides for a disk fixed to the shaft to engage a spring-driven plate attached to the frame.

The internal-expanding brake provides braking action

Fig. 5—Small variable-speed transmissions are now combined with universal motors, giving a compact and efficient assembly which does not have the torque loss or high maintenance of other units

—Photo, courtesy Graham Transmissions

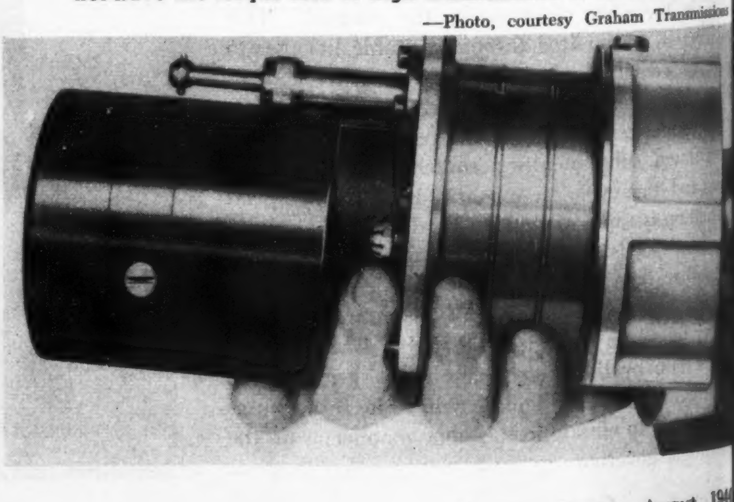


Fig. 6—Right—Aerial-camera motor showing magnetic clutch and brake and spur-gear unit

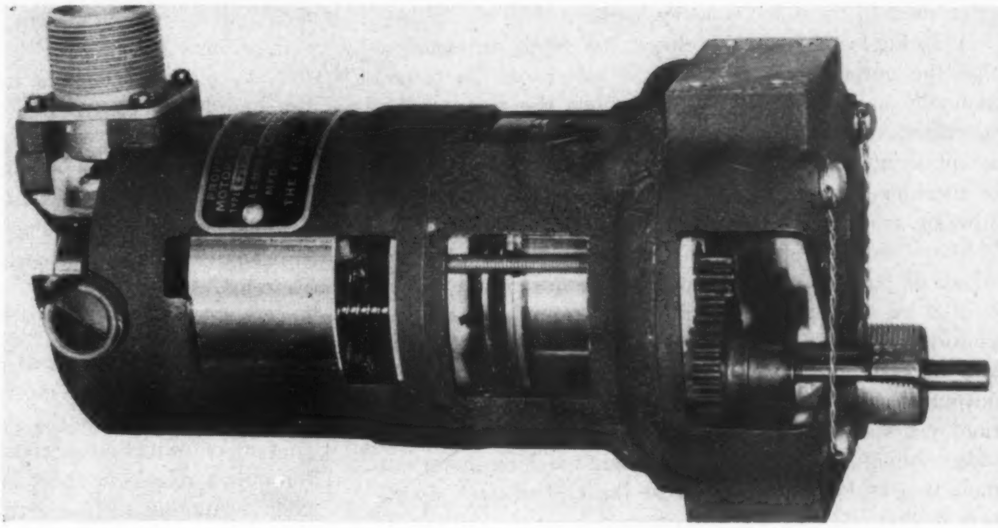
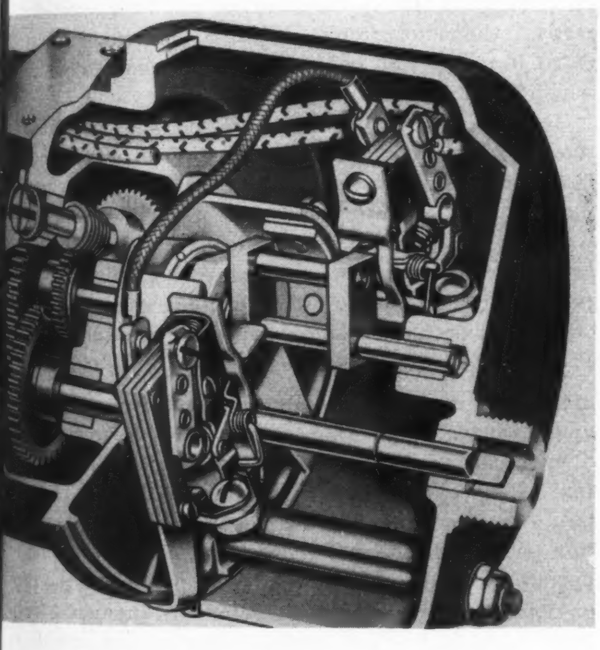


Fig. 7—Below—Cutaway view of aircraft cowl-flap motor with double-reduction spur gearing and limit switches



through contact with the inner surface of a brake drum fixed to the armature shaft. This type generally is shorter than an equivalent disk brake but slightly larger in diameter. The third type of mechanical brake, external-contracting, makes use of shoes which engage the periphery of a drum. This type is larger than either of the other types—disk or internal expanding.

Metal-to-metal contact between the disk or shoes and the braking surface is sometimes used, but the problem of combining a relatively high coefficient of friction with long brake life is best met through the use of some brake lining material such as asbestos or cork. Cork pads riveted to the disk or shoes have proved thoroughly satisfactory in guelling aircraft service. Quick stopping, within about 20 to 40 revolutions of the armature from a speed of 10,000 revolutions per minute, can be obtained from small brakes. For example, one such brake built into a 1/20-horsepower intermittent-duty motor increased the overall motor length by a mere ¾-inch.

The possibilities opened to peacetime machine design by the compact magnetic brake are stimulating to the imagination. For example, a coin-operated vending machine with a precise operating cycle would be a natural application. A portable motion picture projector might well employ the quick-stopping principle to the still-viewing of single frames. Rapid braking between operations would speed up the operating cycles of many light machine tools.

CLUTCHES: The clutch is one of the most versatile and useful adjuncts of a fractional-horsepower motor. Some machines have a low-inertia or high-friction characteristic adequate to the provision of quick stoppage when the momentum of the motor armature is eliminated by disengaging the armature from the output shaft. On high-inertia machines a clutch is frequently used in conjunction with a brake to produce more rapid stopping.

A clutch may also be used to stop actuation of one portion of a device while operation of another portion is continued. Freewheeling clutches permit manual operation of machines driven by motors through irreversible gearing. Still another use of the clutch is to permit the armature to reach full speed before absorbing a high starting-torque load. (Slip clutches will be discussed in Part II of this series under the heading of Protection.)

Of the two main types of clutches the electromagnetic is most commonly used in small motors. The principle is the same as that of the magnetic brake, with the driving surface fixed to the armature shaft and the magnetically-operated driven surface mounted on the driven shaft. These surfaces, frequently cork-lined, are usually disks, the one on the driven shaft driving through a spline which permits it to move axially in response to the impulses of the magnet. The clutch is so arranged that the magnet is energized when current flows in the motor windings and disengagement is accomplished by springs. A clutch-brake combination on a 1/6-horsepower motor stops the driven shaft in 12 revolutions from 10,000 revolutions per minute. Magnet windings generally are compounded to insure holding under all motor loads. Instead of disks, toothed or ribbed clutch surfaces are sometimes employed where clutch slippage cannot be tolerated.

One mechanical clutch design makes use of weighted clutch shoes movably fastened to the armature shaft and restrained by a garter spring. When centrifugal force overcomes spring pressure the shoes engage the inner surface of a drum fixed to the driven shaft. This is the type

often used to break loose heavy loads.

A spring-loaded ratchet clutch has teeth arranged so that the output shaft of a geared motor can be turned manually in the driving direction when the motor is not operating. This type is used in a 1/6-horsepower ammunition booster motor, *Fig.* 2, to permit manual loading of machine-gun ammunition belts otherwise made impossible by gearing.

Many an interesting possibility will occur to the designer of postwar machines when he considers the potentialities of the built-in clutch. For example, a motion-picture projector motor can drive both the mechanism and its cooling fan, with provision for disengaging the output shaft by unclutching to stop the projector operation, yet leaving its fan engaged and operating to provide cooling air. Again, a calculating machine motor can include a clutch to disengage at the end of each operation so that the cam will not overrun.

SWITCHES: The electric switch is, perhaps, the most familiar of all motor control devices, yet its use as a part of the controlled motor has been limited pretty much to portable tools and machines requiring simple on-off or reversing switch control.

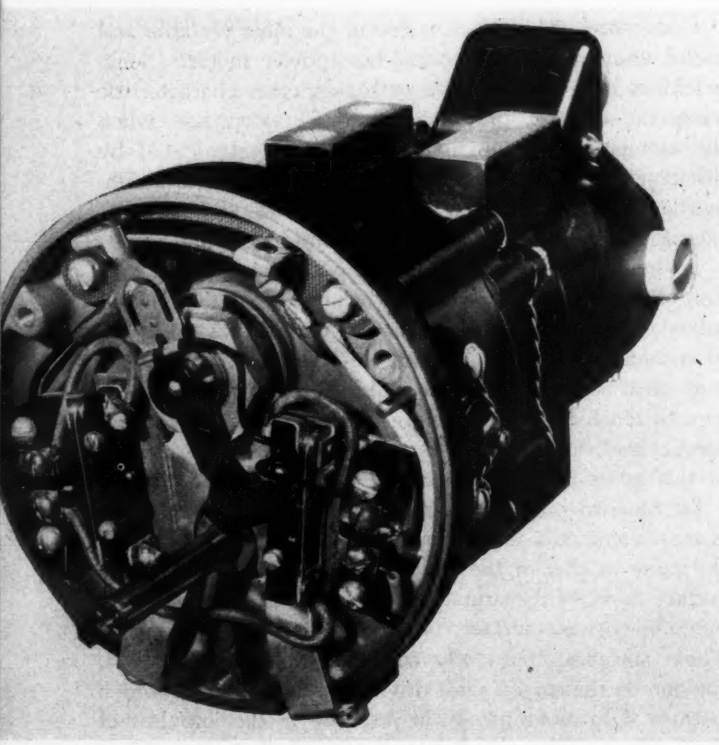


Fig. 8—Limit switch mechanism employed in aircraft cowl-flap motor

In many machines, to be sure, the motor is so located that the on-off switch cannot be built into it. There are, however, other ways in which switches can be used to increase the utility of the motor and reduce external complications; for instance, automatic control (which will be discussed later) and limit switches.

The purpose of the limit switch is to open the motor circuit when the driven mechanism reaches its limit of travel. Scores of devices operate for a fixed number of revolutions or seconds, or a member travels an established distance. For such applications, motor switches can be arranged to trip at the end of the cycle. This technique is similar to the use of stops on a lathe to limit the length of cut. Small switches with relatively high current-interrupting capacity now are available, lending themselves readily to installation in motors. Their operation in conjunction with motor-actuated cowl flaps, wing flaps, and similar mechanisms has proved extremely dependable—even at high altitudes.

Cams Operate Limit Switches

Of the various methods of associating switches with the output shaft, that most frequently encountered employs a cam geared to the shaft, *Figs.* 7 and 8. Since the amount of reduction is considerable, the cam itself turns through a relatively short arc, usually less than 180 degrees. Another technique utilizes a screw and travelling nut—to drive the switch-operating lever through a yoke. In most small motors the switches can be connected directly in the motor circuit. Where motor current is heavy, external relays can be used with the built-in switches handling only control current.

To take care of variations in the machine, internal screws or external levers are provided to afford an adjustment for limit switches. The surest way to prevent recycling with cam-operated switches or switch damage with lever-operated switches is to incorporate a brake in the motor design. Motors not employing a brake for this purpose ordinarily make use of long-lobe cams or switches with provision for substantial overtravel as a safety factor.

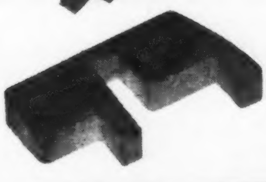
AUTOMATIC CONTROLS: Several aircraft accessories are used to control temperature. Cowl flaps regulate engine cylinder temperature, oil-cooler doors regulate oil temperature, while intercooler doors regulate the temperature of supercharged air before it is fed to the carburetors. In actual practice the pilot observes the temperature on an indicator before him and presses a button to operate the motor which adjusts the position of the flaps or doors regulating the temperature in question. Obviously a method which relates motor control directly to temperature would relieve the pilot of several duties and provide more accurate temperature regulation. After a period of development, automatic motor controls have become available—controls responsive to temperature, pressure or speed. As yet, however, the control unit cannot be built into the motor conveniently but is compact enough to be attached to it, making an integral assembly.

Primarily the unit consists of an element such as a bimetallic strip or liquid-filled bulb which is sensitive to the condition to be controlled, and a system of relays or switches which translate impulses from the sensitive detector into motor operation of the regulating mechanism. The detector of course may be located remotely from the motor with the impulses transmitted by wires or a tube. An important feature of the unit must be a so-called "dwell period" to prevent extremely frequent cycling or hunting of the motor.

Other types of automatic controls include photoelectric units, time switches, and eventually, perhaps, wireless systems.

Built-in auxiliaries for motion modification, protection and indication will be discussed in Part II of this series to be published next month.

What Brass Powder Parts Offer the Designer



By E. H. Kelton
Development Engineering Division
The New Jersey Zinc Co.

Fig. 1—Made via brass powder metallurgy, the machine parts pictured above demonstrate the range of shapes which can be produced on a mass production basis. (Photos by courtesy of Powder Metallurgy Corp.)

WHEN properly designed and produced, machine parts such as those pictured in *Fig.* 1 may be made of sintered brass powder at considerable savings. However, like all other fabricating processes, brass-powder metallurgy has inherent characteristics which must be understood by the designer before he can utilize the benefits of the process fully.

Brass, one of the most commonly used metals in wrought form, has received little attention in powder form in spite of its desirable properties after sintering. Early trials of brass powder by commercial fabricators indicated that atmospheres generally used in sintering iron and copper powders were not suitable. However, laboratory investigation has revealed several atmospheres which are satisfactory and, accordingly, this factor no longer should impede brass powder development.

ALLOYS: As a generic term, brass covers literally dozens of compositions of copper and zinc, often with one or more other elements (lead, iron, tin, nickel, etc.) added in relatively small quantities. All of these compositions may be fabricated by powder-metallurgy methods. Many variations of straight brass powders have been studied and laboratory tested and some of them have shown unusual properties but, for the sake of clearness, only 70-30 and 90-10 brasses with minor alloy modification will be considered in detail in this article.

For certain purposes, elements are added to wrought brass which make the melting, casting, rolling, and perhaps even the fabrication more difficult and hence increase the cost of the finished parts. In brass-powder metallurgy, however, there are cases where the addition of other powders to brass powder by mechanical mixing causes no significant change in fabricating properties

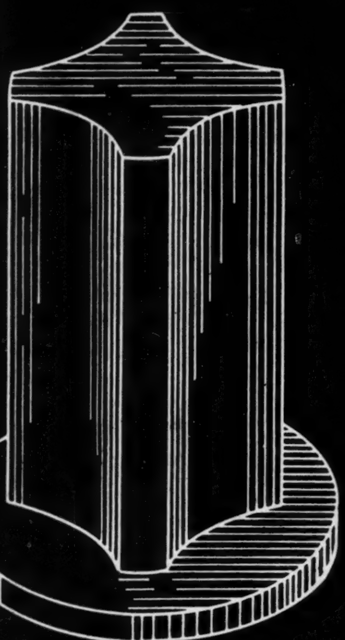


FIG. 2

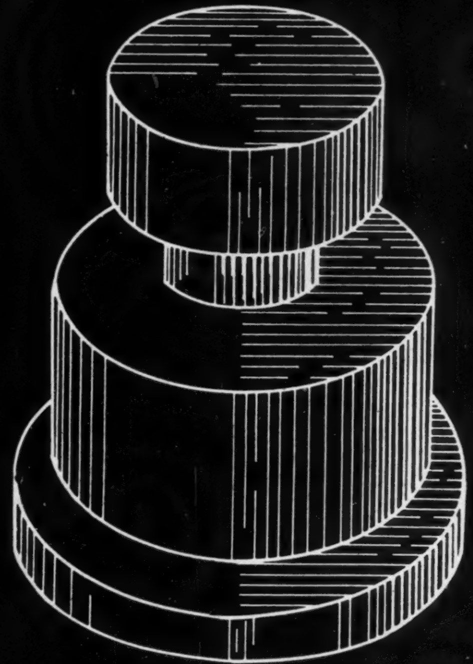


FIG. 3

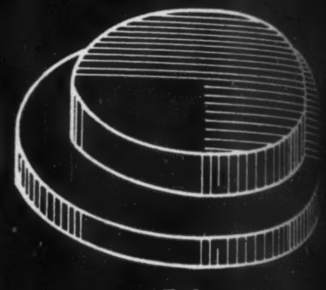


FIG. 4-A

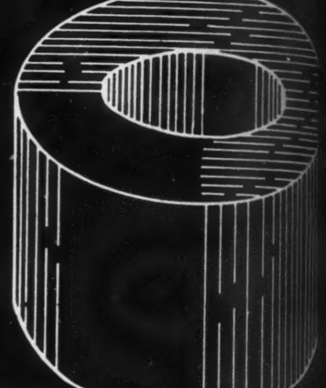


FIG. 5-A

Fig. 2—Most economical are parts of longitudinal section having lateral projections at only one end

Fig. 3—When design permits, such undercuts in the direction of pressure application should be avoided

Figs. 4a and 4b—Short or long sections in the direction of pressure application may be produced

Figs. 5a and 5b—Holes parallel to longitudinal axis can be cored. Others must be drilled after sintering

Fig. 6—Section of least density for part (left) made in single-action die; (right) made in floating die

Fig. 7—Density variations in this type of part can be minimized by proper design of the compacting die

and the troubles of wrought brass processing are avoided.

ATOMIZING: Production of brass powder by atomizing a molten alloy insures homogeneous structure and greatly increases the ease of obtaining excellent properties in the final sintered compact. Inferior results generally are obtained when mixed copper and zinc powders are used for making the compact. However, it is practical to add certain third elements to atomized brass powder.

Although in atomizing brass powder a wide distribution of particle size results, it is feasible to narrow this range by screening or other methods of separation. Some control of particle shape also is possible. For practical purposes, however, the powder particles are of irregular or spherical shape with rather wide size distribution. Only −100 mesh powder will be treated specifically here.

Properties Depend on Many Variables

PROPERTIES: Mechanical properties of sintered brass compacts depend on particle shape, size distribution, surface condition, compacting conditions such as pressure, speed, lubrication, and die design, and on control of sintering atmosphere, temperature, and time. Assuming reasonable control of these factors (which are problems of the powder producer and powder fabricator), the properties shown in the table "Properties of Specimens Made from Brass Powder", may be expected in test specimens as sintered under laboratory conditions. Properties obtainable in commercial articles will fall short of those shown by an amount dependent on shape, tool design, lubricant, and control of sintering.

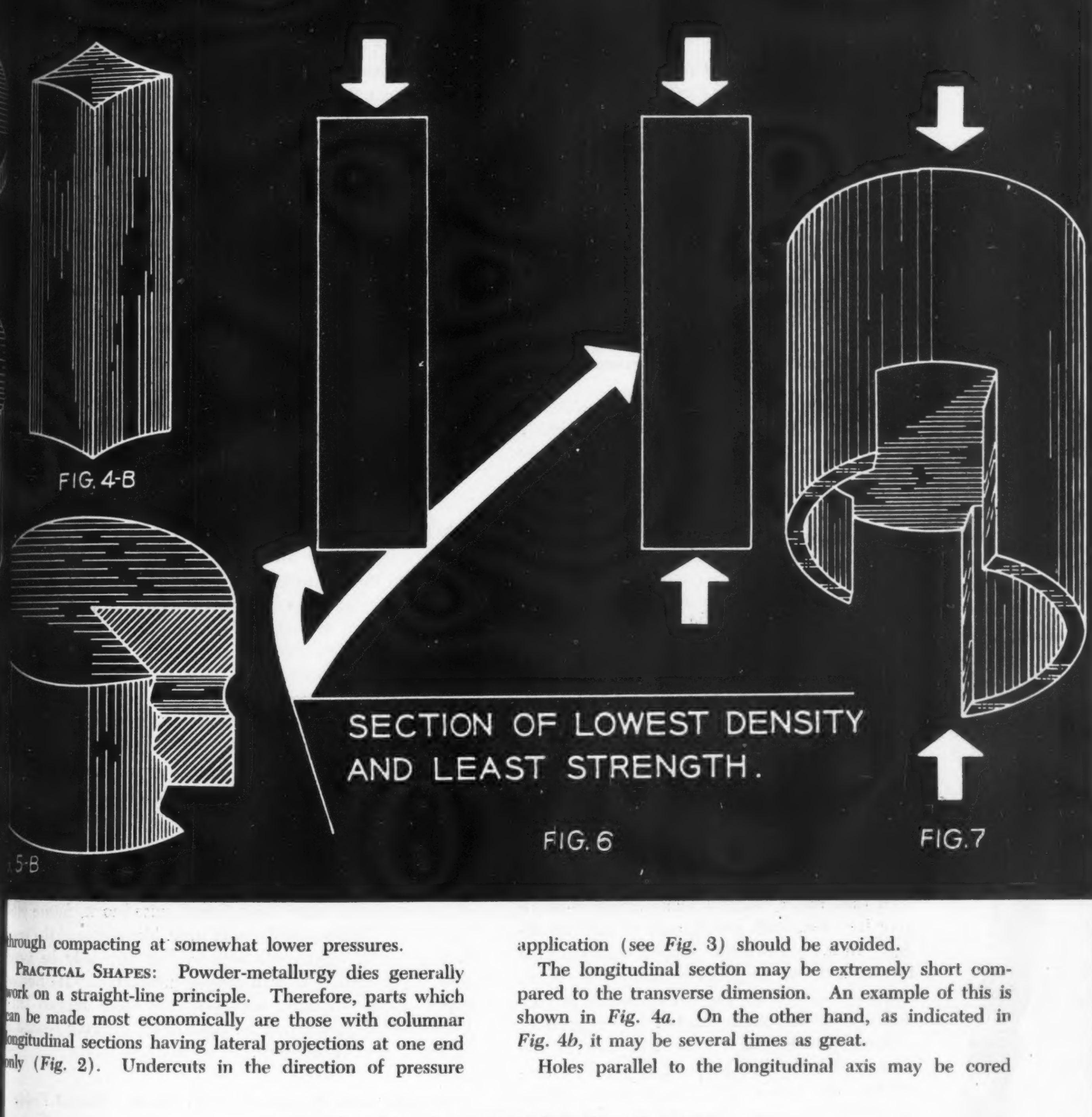
Sintered brass compacts are not subject to aging effects and hence are stable in dimensions and properties. Their corrosion resistance is similar to that of wrought brass.

It will be seen from the properties listed in the table that brass compacts have yield strength and modulus similar to wrought brass and general properties not widely different from those of cast brass.

Where close dimensional tolerances are needed, compacts as sintered often are re-pressed or coined. This increases tensile strength and density and reduces ductility.

Dimensional tolerances to be expected on compacts as sintered are plus or minus .004-inch per inch on the diameter and plus or minus .008-inch per inch in the direction of pressing. Re-pressing or coining will improve the tolerance on diameter to plus or minus .0005-inch but probably will not alter the tolerance in the direction of pressing. Even closer tolerances can be maintained but cost proportionately more.

Re-pressing may be used also to increase density where maximum density is required. On the other hand, when neither maximum density nor maximum strength and ductility are necessary, simplification of fabricating procedure and some economy in metal weight can be realized



through compacting at somewhat lower pressures.

PRACTICAL SHAPES: Powder-metallurgy dies generally work on a straight-line principle. Therefore, parts which can be made most economically are those with columnar longitudinal sections having lateral projections at one end only (*Fig.* 2). Undercuts in the direction of pressure application (see *Fig.* 3) should be avoided.

The longitudinal section may be extremely short compared to the transverse dimension. An example of this is shown in *Fig. 4a*. On the other hand, as indicated in *Fig. 4b*, it may be several times as great.

Holes parallel to the longitudinal axis may be cored

Properties of Specimens Made from Brass Powder

	70-30 Brass				90-10 Brass			
	Specimen A		Specimen B		Specimen A		Specimen B	
Compacting Pressure (tons per sq. in.)	30	50	30	50	30	50	30	50
Sintering Temperature (deg. C.)	900	900	850	850	920	890	910	880
ASTM STANDARD TENSILE BARS								
Yield Point (psi, .2% offset)	15,900	18,800	8,600	8,800	11,500	14,100	9,200	10,300
Elastic Modulus (10^6 psi)	13.2	16.3	11.9	13.9	16	16.7	11.4	12.9
Tensile Strength (psi)	29,100	35,200	32,000	34,900	26,100	28,400	27,400	29,700
Tensile Elongation (% in 1-in.)	11	13	43	50	15	13	31	34
Brinell Hardness No. (10-mm ball, 250 kg load)	43	51	35	39	38	49	35	41
Density (grams per cm^3)	....	7.92	......	7.94	....	8.36	....	7.99
SPECIMENS ¼-IN. SQ., 3-IN. LG.								
Shear Strength (psi)	28,800	32,300	25,600	26,000	24,800	25,000	23,500	24,500
Charpy Impact (unnotched, ft.-lb.)	11	13	22*	25*	11	13	25*	27*
Thermal Expansion (10^{-6}mm per deg C)								
25 to 100 deg C	17.7	18.2	18.4	18	17.1	16.9	17.1	17.4
25 to 300 deg C	19.3	19.1	19.6	19.5	17.9	18	18.4	18
SPECIMENS ⅜ BY ¼-IN. BY 3-IN. LG.								
Electrical Conductivity at 25 deg C (% of Int'l Annl'd. Copper Std.)	22.11	25.17	17.19	19.06	33.24	35.72	14.35	16.51
SPECIMENS ½-IN. DIAM. BY .45-IN. LG.								
Compressive Strength (psi, 10% compression)	38,300	43,200	29,500	31,600	34,600	39,000	31,400	32,700

*Pulled through without fracture.

(*Fig. 5a*), but those not parallel must be machined after sintering (*Fig. 5b*).

Brass-powder parts having optimum, uniform properties are difficult to produce when:

1. The pressure axis is great with respect to the transverse axis. More and more of the external pressure applied on the powder is absorbed in die friction and powder particle friction as the distance from the surface increases. Thus, as shown in Fig. 6, when a single-action die is used, density and strength decrease as distance from point of pressure application increases. If a floating die is used, then density and strength will be poorest at the center.
2. Sections of different length in the direction of compacting exist. If the part shown in *Fig.* 7 is round in cross section so that it cannot be compacted except in the direction of the arrows, and a single bottom punch is used, then the densities in the two legs will be less than that in the center portion. In addition, the different sections will have unequal shrinkage in sintering and distortion will result. However, these disadvantages can be overcome by using three lower punches, each having a different length.

Fig. 8—Left—Demonstrating the ductility of ¼-inch square sintered bars of 70-30 brass powder. Can be bent through 180 degrees or subjected to a 720-degree twist

A more usual expedient in difficult fabricating operations is to subject the part to re-pressing or coining after sintering. This offers the added advantage that lower original compacting pressure may be used, with less danger of tool breakage. In many cases the compacting tools may be utilized for re-pressing but when cored holes or cavities are present it usually is necessary to install other punches for re-pressing.

Some powder fabricators feel that in the present state of development of powder metallurgy, the largest field lies in those articles in which optimum properties are not necessary for the utility of the part; i.e., in parts which can be compacted at the lowest pressure providing adequate green strength and which can be re-pressed after sintering when close dimensional tolerance is desired.

Sizes: Brass powder compacts, in order to have sufficient green strength to be handled, should be pressed at 25 tons per square inch or higher. Fabricating difficulties increase rapidly when unit pressures of much greater than 50 tons per square inch are used. Therefore, the size (cross section normal to direction of pressure application) of a part to be considered for powder metallurgy is limited only by the size of the press available, bearing in mind the unit-pressure limits stipulated. Length in the direction of pressure application is limited by design factors already discussed and by the press stroke. Most powders have a compression ratio in the order of 2.5 to 1. Press strokes vary widely depending on type but rarely exceed 8 inches except in hydraulic machines.

Economies which may result from the utilization of brass-powder parts include: Elimination of machining; reduction of inventory stock; elimination of expensive metallurgical processing; reduction in weight; and reduction of scrap loss.

In the final analysis, the decision to use brass powder must be based on one or both of two factors: (1) Economy, (2) some unique feature of brass-powder metallurgy which permits the solution of a particular problem.

Powder Metallurgy Opens New Fields for Copper

POWDER metallurgy appears to have a clear advantage over other processes, regardless of the metal involved, in the production of such items as: Clutch plates or friction metal generally, bearings and bushings (especially porous types), filters, tungsten filaments, Alnico magnets, hard carbide tools, bimetals, contact elements, small gears, copper-carbon brushes, irregular contour cams, and paint pigments.

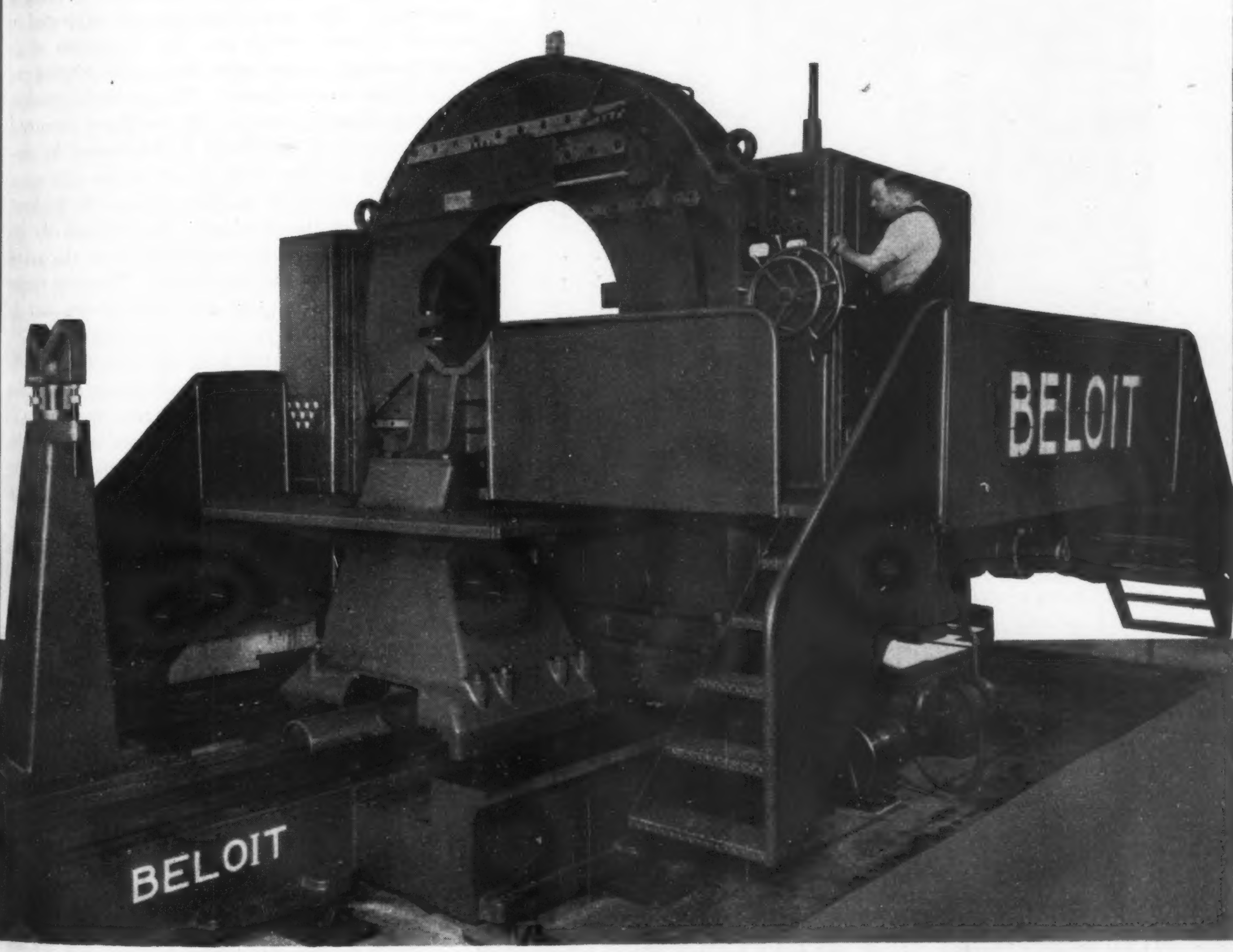
This was the view expressed by Dr. D. K. Crampton at a recent meeting of the Metal Powder Association. After stating that the 1943 total copper powder production (7500 tons) amounted to slightly less than one-half per cent of the 1941 total production of other primary forms of copper, Dr. Crampton remarked that while it is exceedingly difficult to give accurate predictions, from the nature of the powder metallurgy process and the type of articles which have been found suited to it, future production of copper powder will most likely exceed the one-half per cent figure and will in fact probably be more than 1 per cent of overall copper production. He further expressed the thought, however, that there appears little likelihood of copper powder production growing beyond a figure of 2 or 3 per cent.

Wrought and Powdered Forms Serve Different Fields

Of all the items for which powder metallurgy is best suited, Dr. Crampton feels that it is primarily in bearings and bushings that the process has made serious inroads in the fields covered by cast or wrought methods. In his opinion, the great preponderance of things made from wrought copper alloys do not lend themselves well to the powder metallurgy procedure. On the other hand, the articles which currently are made from copper or copper alloy powders are for the most part ones which have not heretofore been made by wrought procedures.

Referring to the attitude of men in the wrought copper industry toward powder metallurgy, Dr. Crampton said: "It is my opinion that powder metallurgy of copper certainly cannot be ignored, and I am equally certain there is no sense in fighting it. Therefore, the wrought copper industry might just as well embrace it and the probability is they will gain more than they will lose by so doing".

Redesigning To Improve Speed and Accuracy



By E. D. Beachler
Beloit Iron Works

UNTIL the war there had not been any crankpin lathes made in this country on a commercial basis but there were a few European-built machines in operation. With the desirability of this type of machine established by the few in use, development was started on a machine patterned after the European type. This lathe, however, had several defects. In redesigning, the gear which drives the cutting ring was enclosed, improved lubrication was installed on both the main drive gearbox and the longitudinal feed transmission gearbox, and better controls were provided the operator for "jogging" and controlling both the longitudinal and ring-drive motors.

The result of this design is shown in *Fig.* 1. Since that time, still further improvements have been incorporated which will be discussed in detail later.

The principle of these machines is best under-

Fig. 1—Top—Crankpin turning lathe achieves high production and precise operation with highly automatic controls and novel design features

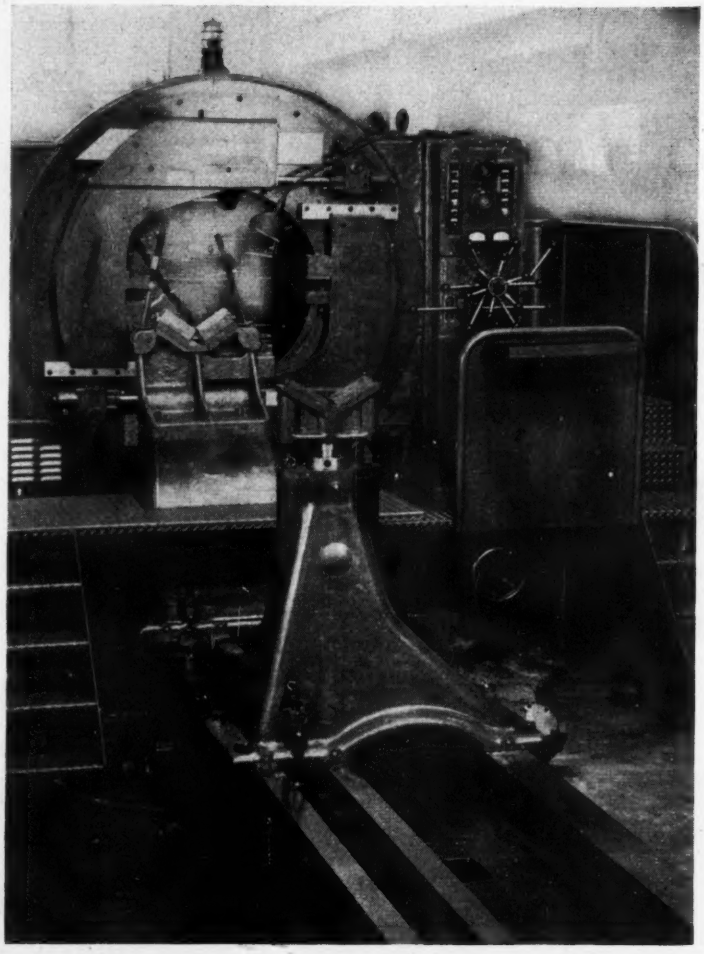
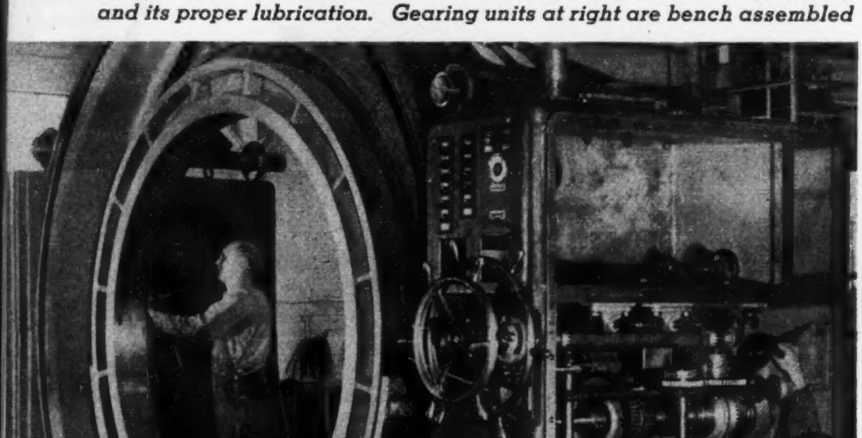


Fig. 2—Above—Trunnions for holding crankshaft in position have chilled cast-iron pads, giving better wear service than hardened steel

Fig. 3—Below—Accuracy of machine depends upon main ring bearing and its proper lubrication. Gearing units at right are bench assembled



stood by comparison with a conventional engine lathe. In a conventional lathe it is necessary to set up the crankshaft off center on the center line of the pin diameter. Then this large mass is rotated off center to turn the cheeks, heels and pin diameter of the crankshaft. This means that cutting speeds are low and the amount of feed is small because of the difficulty of supporting the work rigidly enough to stand heavy cuts.

Tools Travel Around Crankshaft

In contrast with that, the crankpin turning lathe supports the crankshaft in a stationary position on stanchions. The main bearings are supported by sets of V-blocks which put the center line of the main bearings on the center line of the rotating ring when throw is set at zero. The particular crankpin to be machined is set up with the throw horizontal. Exact amount of the throw is determined by moving the ring housing with respect to the carriage on the bed in a direction at right angles with the longitudinal axis of the machine. To machine the pin itself, the crankpin is then located on the center line of rotation of the face plate. The ring rotates around the crankpin with the tools moving and the work being stationary.

By means of a radial feed, the two tools on the tool slides can be moved in and out to the cutting diameter as required by the particular operation. To machine the length of the pin and the distance across the heels, the carriage and ring housing assembly are moved longitudinally on the main bed. Additional steady rests support the overhung ends of the crankshaft in addition to the two stanchions which are close to the work being machined.

When one pin is completely machined, either the carriage is moved along on its bed to machine the second pin, or the crankshaft is shifted in the stanchions to the new position to accommodate the next pin. After the first pin is machined, a protractor is attached to the crankshaft and remains until all the pins have been machined. This protractor is provided with an accurate machinist's level and is graduated in degrees. The first throw is set on zero. Next, the operator moves the protractor through a suitable number of degrees to give the difference between the first and second pins to be machined. Then the crankshaft is rotated until the level reads zero again, putting the pin in its correct position.

Taking up the detail parts of the machine, the bed has flat ways and heavy construction. These ways are flat for ease in machining and scraping to a level surface. The bed was designed so that it would accommodate the main ring housing carriage and at the same time provide for the necessary stanchions. In the larger sizes of the machine, the bed was extended so that the outboard steady rests also are accommodated on a reduced section of the main bed. In the case of the preced-

ing lathes, extension floor plates were used for the outboard steady rests and were grouted directly into the concrete. The stanchions supporting the work are of a design to clamp the main bearing journals adequately during the various machining operations. By comparing the stanchions in *Figs.* 1 and 2, it can be seen that special spacer blocks have been added in *Fig.* 2. These blocks are changed to accommodate various shaft diameters and are a special chilled cast iron in place of hardened steel which was used previously. The new blocks have been found to stand up much better under the wear and abuse received in holding and changing the crankshaft from position to position.

The main carriage is fitted with adjustable gibs which bear against the sides of the center of the main bed. These gibs are close together to keep the carriage in alignment as it moves longitudinally. The carriage itself is provided on the outside ends with adjustable gibs to obtain the proper clearance and, in the case of the smaller models, to prevent lifting of the ring housing during machining operations. In the case of the largest machines, the weight of this housing is sufficient to prevent any lifting since the machines weigh between 225,000 and 250,000 pounds total.

Does Not Affect Alignment

Moving parts on the carriage that come in contact with the bed are provided with lubrication from a pump which moves with the carriage. This particular pump operates whenever the carriage is moved longitudinally in rapid traverse. The nature of the lubricant is such that it is difficult to wipe off and will maintain a film of lubrication with a small supply. This is important since it is not possible to collect the lubricant again and recirculate it due to the nature of the machine. This lubricant is not supplied during a longitudinal feed cut because it is important that the oil pressure should not build up enough actually to lift the carriage and destroy the axial accuracy of the ring housing assembly.

To set the stroke, a motor and handwheel adjustment move the ring housing on the carriage. Both the motor, which is used for cross traverse, and the handwheel, which is used for fine adjustment, are connected to a worm gear and screw assembly which move the ring housing with the nut secured to the ring housing base. Provision is made so that only the motor or the handwheel is connected to the worm unit. When the motor is connected, the handwheel is disconnected so that the operator could not be injured by having the handwheel driven by the motor. If the handwheel is connected, the motor is disconnected so that, here again, the operator could not be injured if the motor were accidently started. The push-button control for the cross traverse motor is located on the carriage, close to the handwheel for the convenience of the operator.

In the latest machines the ring housing appears symmetrical. The necessary gearing and drive equipment is located on the right-hand side facing the ring housing and the opposite compartment is used for the electrical and lubricating equipment for the main-ring housing. Oil equipment consists of an oil pump and large filters together with the necessary pressure regulators and pressure gages to control the flow of oil. Oil is supplied to several points on the main-ring bearing which is shown in *Fig.* 3. At the left-hand center of this photograph are a number of pressure gages which indicate the oil

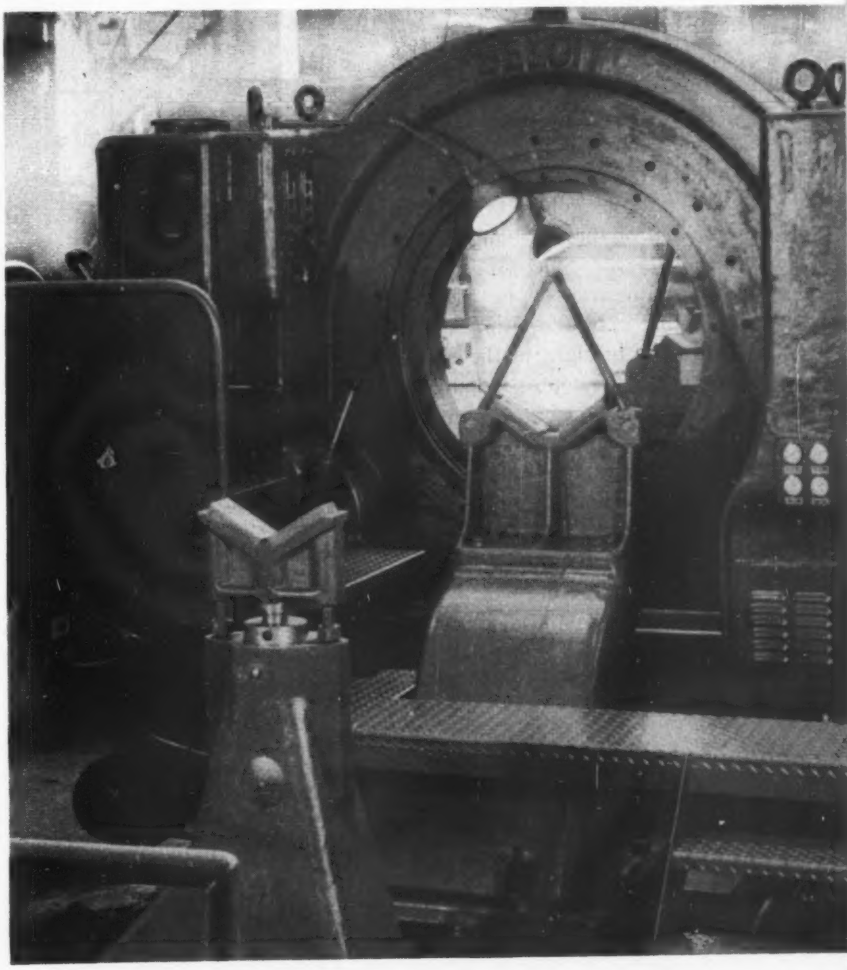
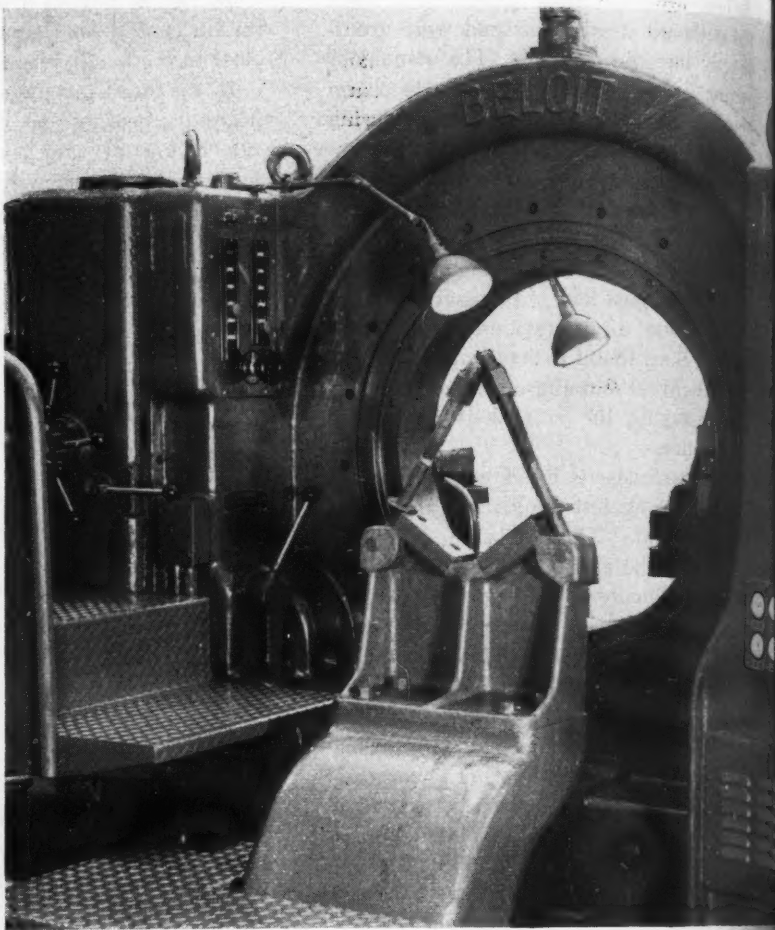


Fig. 4—Main-drive motor is flange mounted to ring housing as shown in this rear view of the main-ring carriage

pressure before and after the filters, in addition to the pressure in the lines supplying the main bearing and gearbox bearings and gears. By reading the differential across the filters, the operator can determine the proper time to clean the oil filters.

Particular care was taken in the design of the main bearings and their lubrication so that oil is supplied at the correct points to maintain the proper oil film. Since this main ring bearing is the "heart" of the machine, it is particularly important that the accuracy built into it can be maintained. It has to take a radial load, thrust in both directions at times and also a vertical lift load during certain portions of the ring revolution. The accuracy with which a crankpin can be turned is no better than the accuracy of this bearing. Since it is larger in diameter than the hole in the ring, every care was taken to

Fig. 5—Duplicate operator's panel is installed on back side of smaller machine



see that proper lubrication was obtained. Crankpins as large as 20 inches in diameter have been turned on these machines with less than .001-inch out of round.

To prevent chips and dirt getting into the inside of the ring housing, special seals are built into the rotating parts of the face plate. All around this large plate and the corresponding one on the back, combination felt and labyrinth seals are used to keep the chips and dirt out.

The ring is driven by a combined variable-voltage motor field-control electric drive. In the case of the large machines, a motor generator set is mounted in the top of the left-hand compartment on the ring. Since the ring housing moves with respect to the floor, this means that only alternating-current power lines are brought to the ring housing. The main drive motor is flange mounted on to the ring housing as shown in *Fig.* 4.

In the gearbox shown in *Fig.* 3, double reduction gearing is used to connect the motor to the ring face plate. To control the ring itself, pushbuttons are located on the operator's panel. In the case of the large lathe, an additional pendant pushbutton is used which can be swung to the position of the operator whether he is working on the front side or the back side of the machine. On the smaller machine, a second operator's panel is installed on the back side as shown in *Fig.* 5.

Pushbuttons Are Located at Each Station

With these controls, the operator can "start", "stop", "inch forward" or "inch reverse" the ring face plate. On the same panels, other controls operate the radial rapid traverse motor, the longitudinal rapid traverse motor and the longitudinal direct-current feed motor when used. Duplicate pushbuttons are provided at each operator's location. The controls are interlocked so that if oil pressure is not available to the main-ring bearing and gears, it is impossible to start the ring face plate. Red and green indicating lights show that lubrication is either "on" or "off".

Main difference between the original lathes and the latest types is in the manner of radial feed. On the first lathe, a star feed type of radial feed was used. On the last, a power means of radial feed has been incorporated. The operator can feed the tool in or out at a definite feed per revolution, he can feed it in or out by hand at any time with the ring stopped or running, or he can feed it in or out with the radial rapid-traverse motor.

In *Figs.* 1 and 2 is shown the method of mounting the tool holder on slides which are bolted to the face of the ring face plate. The tool holder is provided with a nut which works on a radial feed screw going from the tool holder into a gearbox mounted on the face plate. The screw is driven by a worm gear assembly which, in turn, is driven by a spur gear pinion. Whenever this pinion rotates, the radial feed screw is turned which, in turn, moves the tool holder in or out on the cross slide. In the last design, a slip clutch has been incorporated between the worm gear and radial feed screw. This clutch provides protection in case the tool holder reaches either end of its travel or in case the tool is jammed against the work.

The pinion on the worm shaft of this gearbox is meshed with a large diameter gear mounted on the face plate. This large gear is free to turn on the face plate in either direction. If this large gear is stationary with respect to the face plate, there will be no radial feed. If it moves faster than the face plate, the radial feed screw will be turned in one direction. If it moves slower than the face plate, the radial feed screw will be turned in the opposite direction. *Fig.* 6 shows an elementary diagram of the gearing arrangement.

Differentials Provide Feeds

To drive the large diameter feed gear at the same speed as the ring at all times, power is taken off the main ring drive gear train and through a differential which, in turn, is used to drive the large diameter feed gear. The ratios and number of gears in the train are selected so that when the center part of the differential is stationary, the large diameter feed gear is driven at the same speed as the ring. If the center part of the differential is rotated in one direc-

tion, it will make the feed gear rotate faster than the ring; if the center part of the differential is rotated in the opposite direction, it will make the feed gear rotate slower than the ring. In this way, the tool holder can be fed in or out in a radial direction.

Three methods of driving the center part of the feed differential are used. In the first case, in order to obtain a definite feed in thousandths of an inch per revolution, power is taken from the main drive gear train through a series of shifting gears. These gears are laid out so that eight different output ratios are obtained. The output from this transmission then goes through a reversing clutch to reverse the direction of rotation. From that reversing clutch, power is fed through another differential. The input shaft is connected to one side of the differential by means of a worm gear.

The other side of the differential is driven by the radial rapid-traverse motor which normally is stationary when under power feed. Either half will drive the center part of the differential which, in turn, is connected to the center part of the feed differential. The only purpose in using the second differential is to eliminate the necessity for any clutches or disconnect gears to combine the power feed and the rapid traverse motor. With this combination, the operator can be running the tool in under power radial feed and at the same time operate the rapid traverse motor, if he so desires, either in or out without doing any damage to the equipment.

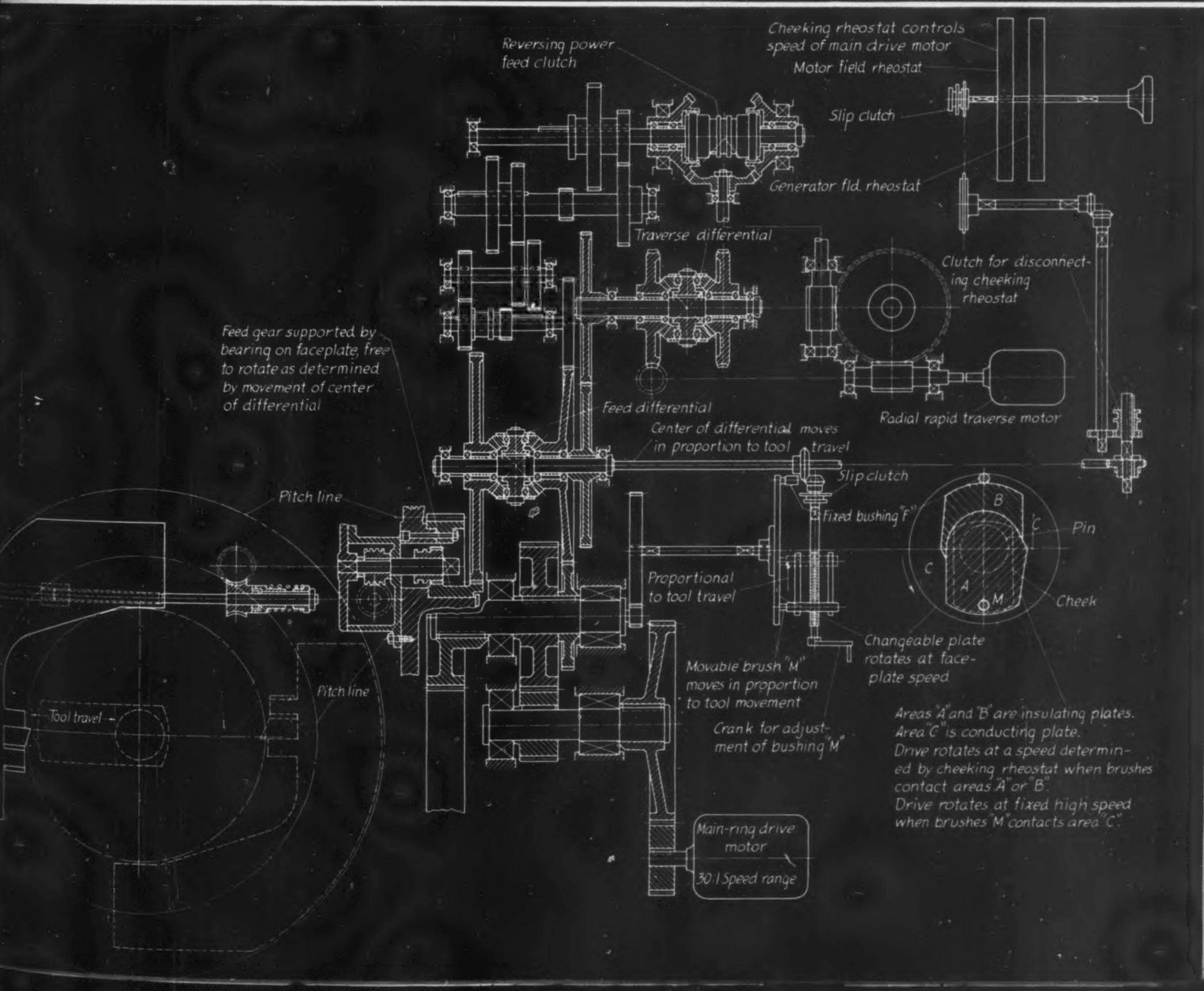
The third method of driving the radial feed is by a handwheel. If the power radial feed is in neutral, it is possible to engage the handwheel gear into the train of gears leading to the power side of the second differential. Here again, the gearing is interlocked so that it is not possible to operate the radial feed under power in case the handwheel is connected. It would be possible to use the radial rapid-traverse motor if the operator so desired, which is unlikely. If the rapid traverse motor were started it could not hurt the operator.

Rheostat Step-up for Intermittent Cuts

When the cheeks of the crankpin are being machined, the cutting diameter of the tool varies widely, sometimes as much as 4 or 5 to 1. If the revolutions per minute of the main ring were left constant, based on the best cutting speed at the maximum diameter, the speed of the cutting tool in feet per minute at the small diameter would be very slow. To eliminate this loss in production due to too slow a cutting speed, the center part of the radial-feed differential drives a special rheostat which, in turn, controls the motor and generator of the main-ring drive.

The ratios are laid out and the rheostat designed so that

Fig. 6—Elementary gearing diagram showing use of differentials for feed and traverse as well as automatic speed-up drive for intermittent cuts



as the tool feeds in, movement of the cheeking rheostat will increase the speed of the ring-drive motor so that essentially constant cutting speed is maintained. The ratios in this gear train are selected so that the rheostat will be moved over the proper portion of its travel, depending on the initial cutting speed which is selected by the operator.

In *Fig.* 7 are shown the location of the various control levers. Levers labelled "A", "B" and "C" control the sliding gears in the transmission for the power radial feed. By properly selecting the lever positions, radial feeds from .008 to .240-inch per revolution can be obtained. Amount of feed is laid out so as to increase in geometric progression. The lever underneath the radial feed nameplate is used for interlocking the radial feed handwheel and the power feed. When this lever is thrown so that the radial feed handwheel can be engaged, the smaller of the two

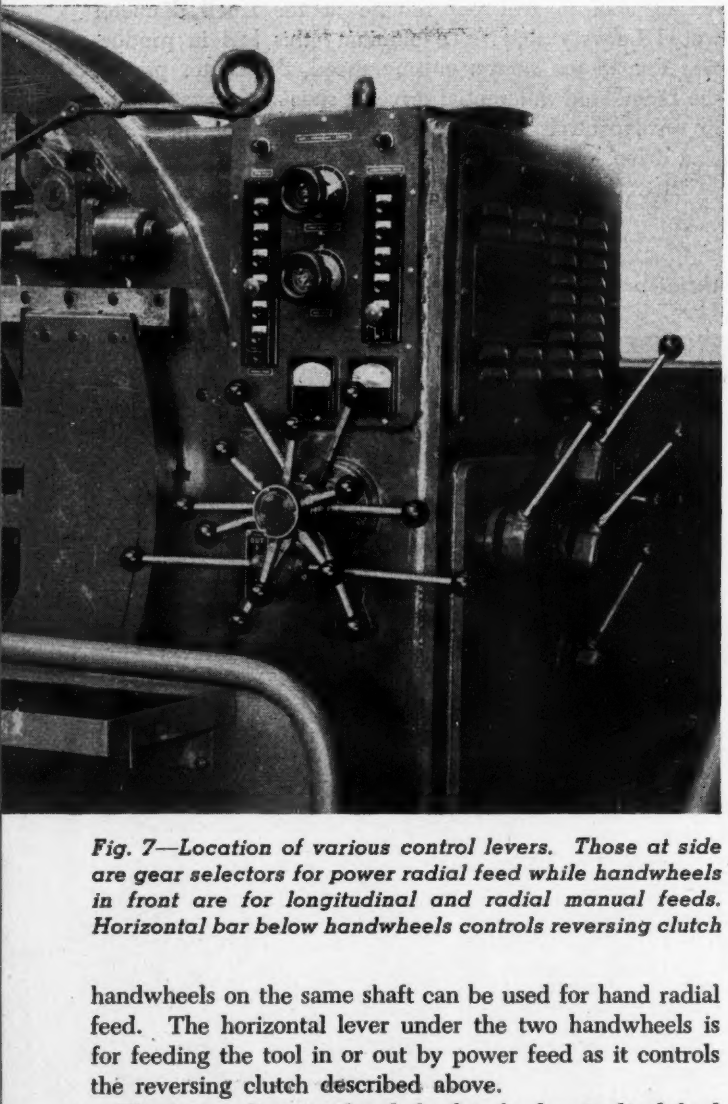


Fig. 7—Location of various control levers. Those at side are gear selectors for power radial feed while handwheels in front are for longitudinal and radial manual feeds. Horizontal bar below handwheels controls reversing clutch

handwheels on the same shaft can be used for hand radial feed. The horizontal lever under the two handwheels is for feeding the tool in or out by power feed as it controls the reversing clutch described above.

The larger diameter handwheel is for longitudinal feed of the carriage. A train of gears is connected from this handwheel to a nut on the longitudinal feed screw. Whenever the handwheel is rotated it will, in turn, rotate the nut and feed the carriage longitudinally.

On the 40-inch and smaller size lathes, this longitudinal feed screw is turned to give the power longitudinal feed. For obtaining the low rates of feed when cutting, a wide range variable-voltage direct-current drive is used. This drive is controlled by a combined rheostat, part of which is in the generator field and part in the motor field. Push-buttons are provided to run the motor in a direction to feed right or left or to stop it. The actual power from the motor is transmitted through a worm gear to one part of a differential. A rapid-traverse motor is connected to the other half of the differential by means of a worm gear also. The center part of the differential is connected to the feed screw. With this arrangement it is possible to run the longitudinal direct-current feed motor and then to superimpose the rapid-traverse motor in either direction in case the operator so desires in an emergency. On the larger lathes, the longitudinal feed is obtained by the same gears which supply power to the radial feed.

The gear box itself is designed so that each shaft of gearing may be removed readily from the box without disturbing any more of the remaining gears than absolutely necessary. This was accomplished by making small subassemblies with proper mounting provisions in the box itself to keep them in alignment. Most of the bearings are of the antifriction type and adequate lubrication is provided all the bearings and each gear train. By the operator checking the oil-pressure gage mentioned before, he can be assured that the gearbox as well as the main bearing is receiving oil at all times.

Radar Utilizes New Plastic Material

MANY unusual properties are exhibited by polyethylene, a new thermoplastic resin. A high-pressure synthesis product, this thermoplastic is now being employed in radar equipment for insulation of coaxial cable.

By changing the conditions of manufacture, the properties of the base resin can be varied to meet the needs of particular applications. At the present time, however, only the one grade is being produced. Because the resin in inherently flexible the use of plasticizers is unnecessary in processing.

Among the properties which may be varied are tensile strength, elongation, tear resistance and brittleness temperature. Being colorless as manufactured the resin can be produced as translucent or formulated in a wide color range, with exceptionally high luster. The plastics developed by Carbide and Carbon Chemicals Corp. in collaboration with The Linde Air Products Co., are so light that they will float on water, and can be processed by conventional methods.

All types are characterized by a dielectric constant of 2.29 (at 50 megacycles and 25 degrees Cent.), a power factor of .0003, low water absorption (.03 per cent weight gain in 100 hours), low transmission of water vapor, and resistance to attack by many chemicals including a number of solvents. The electrical grade has a specific gravity of .92, specific heat of .53 (between 18 to 40 degrees Cent.), softening temperature (transparency point) of 105 degrees Cent., linear coefficient of expansion of 25×10^{-5} cm/cm/C (above 115 degrees Cent.), tensile strength of 1800 pounds per square inch, elongation of 600 per cent, notch impact strength greater than 3 ft.-lb./in., and brittleness temperature below −70 degrees Cent.

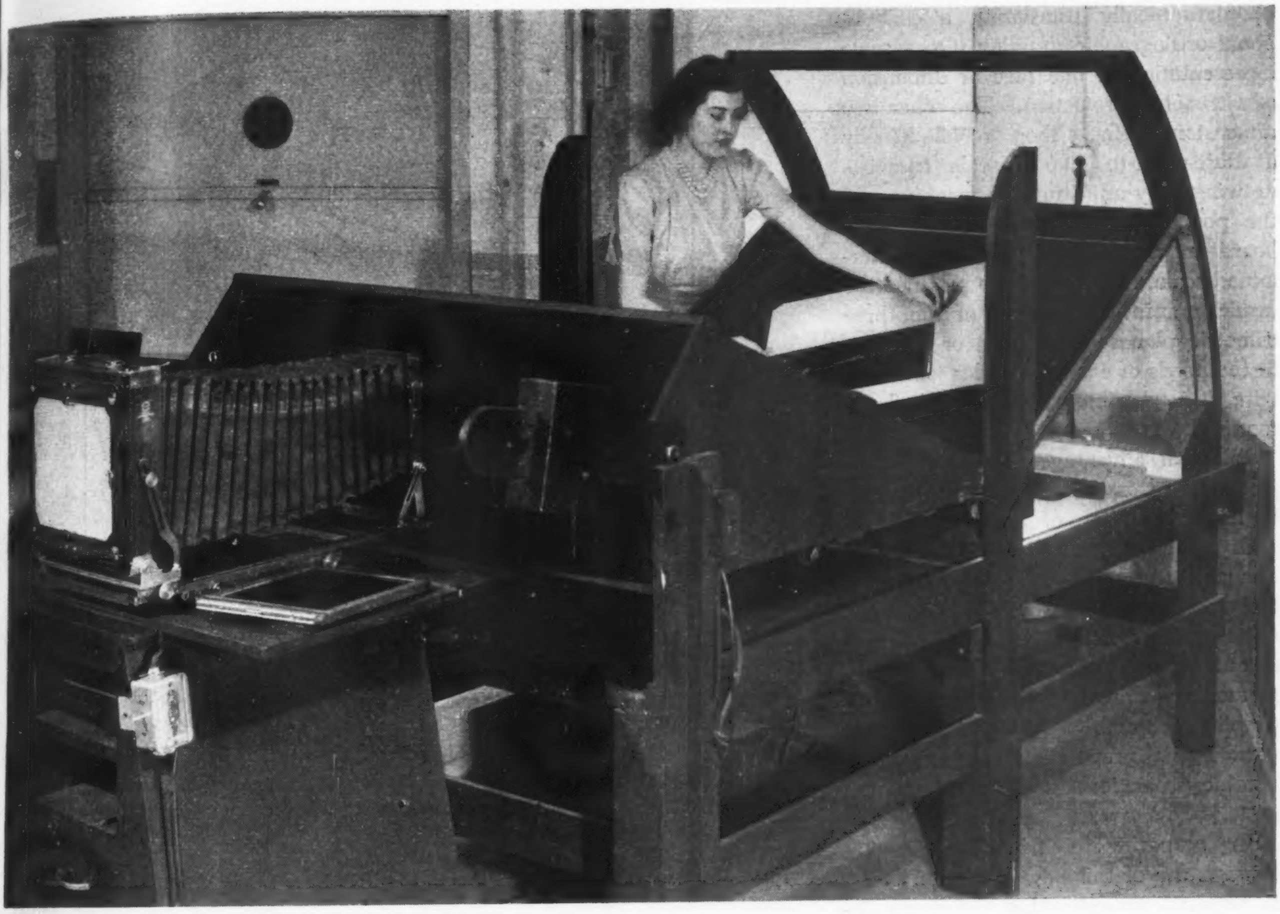


Fig. 1—Above—This machine converts orthographic drawings into trimetric projections. Operator is shown setting drawing to be photographed at appropriate angle

New Trimetric Drawing Technique

By W. C. Wilkinson and H. C. Bartholomew

Engineering Department
The Glenn L. Martin Co.

TRIMETRIC drawing is a type of three-dimensional illustration which has been recognized for some time as being extremely useful in production work. Its infrequent use has been due largely to difficulties involved in preparing the drawings. However, many of these difficulties have been obviated through recent developments made by the authors specifically for use on production work at Glenn L. Martin Co. Prime development was a machine, the "Axonograph", which used in conjunction with four new drawing instruments cuts 50 to 80 per cent off time previously required to prepare a trimetric drawing.

This new machine, *Fig.* 1, is a device for

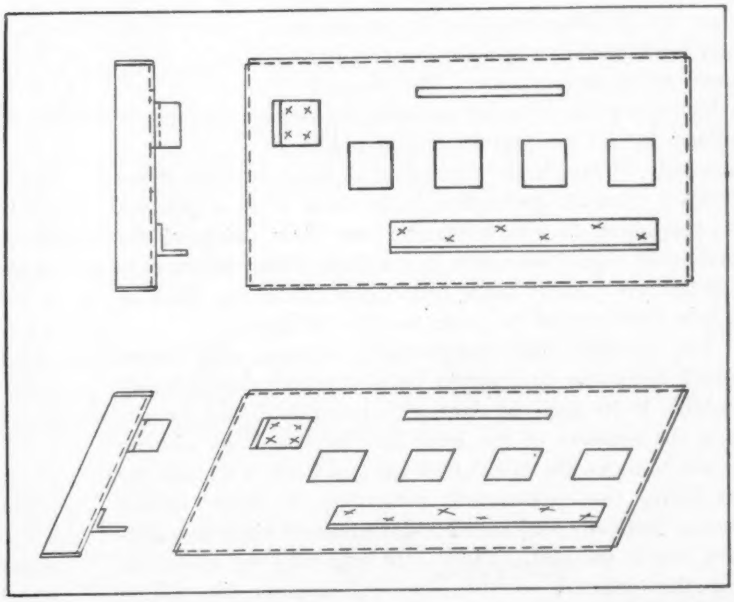


Fig. 2—Top view shows orthographic drawing of part which fits in the plan plane of an assembly. Lower view shows same drawing after it has been converted into a trimetric projection by being photographed at proper angle on the machine pictured in Fig. 1

photographically translating a conventional orthographic drawing into a scale representation of one face or dimension of a trimetric projection. The other two dimensions or faces then are filled in by a draftsman to give a scale trimetric drawing. By adjusting the machine and the position of the print on the copy board in accordance with a predetermined scale, it is possible to produce the resulting print in any one of the three trimetric planes—plan, face or side—to fit the needs of each particular job. Likewise, it is possible to vary the scale as required.

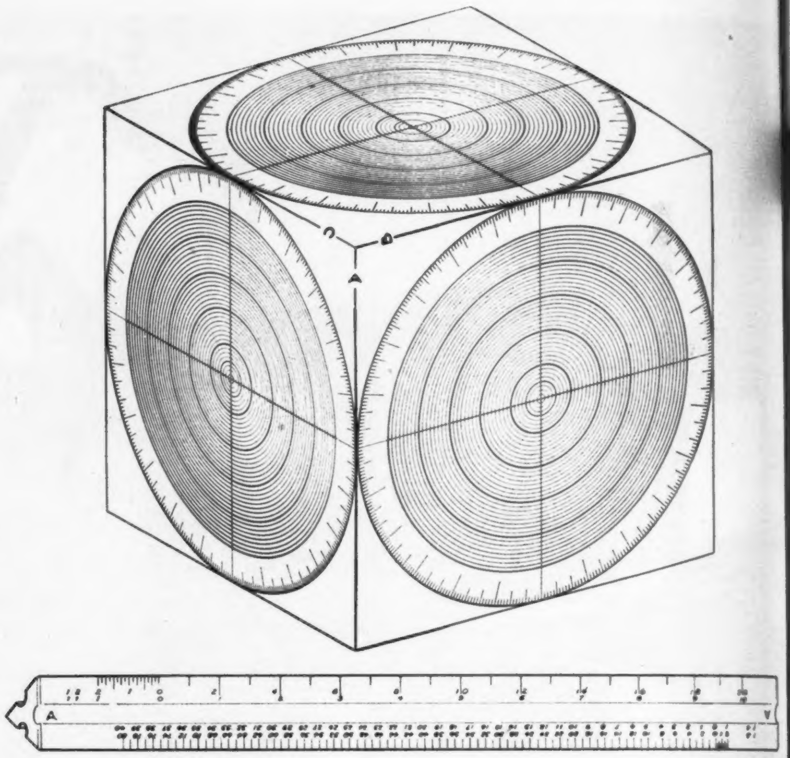
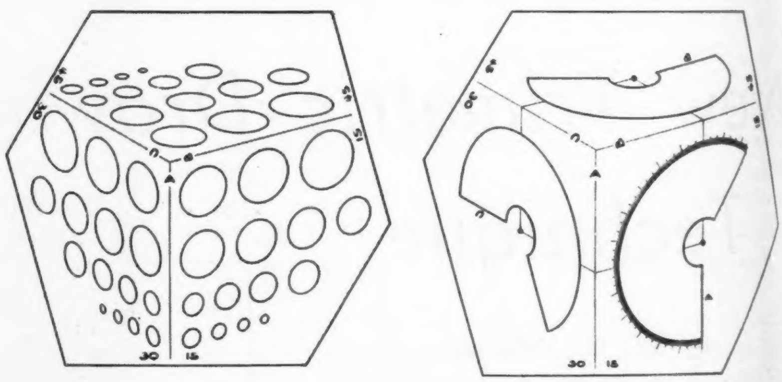


Fig. 3—Above—Underlay at top is used to trace off ellipses on trimetric drawings. Scale is for measuring on the three major trimetric axes

Fig. 4—Below—At left is shown the template used for drawing ellipses. Instrument at right is a trimetric protractor used for measuring angles



Principal advantage of trimetric projection is in assembly work where, due to its presentation of three dimensions with a minimum of distortion, it offers a clear picture of how the various parts and sub units are assembled. It in no sense replaces orthographic projection in manufacturing operations, but is a supplementary drawing which speeds and simplifies assembly work. As such, it normally is prepared after the orthographic projections of the various detail parts.

In preparing a trimetric drawing through use of this new machine, it first is required to determine the scale desired on the finished drawing. Next, the drawing is broken down into its various component parts which are sent to the machine with instructions as to scale and the principal plane in which each part is to appear on the finished trimetric.

If all prints are to the same scale one machine setting only is required, but if they are to different scales the machine can be adjusted to compensate and produce prints to one scale. The plane in which the print is to be made is determined by its position in the complete assembly. Thus, in the case of an airplane fuselage, if the finished trimetric projection is to show it in a position corresponding to a three-quarter rear view, prints of the bulkheads would be made in the face plane, prints of a top hatch would be made in the plan plane and those of a side hatch would be made in the side plane.

The machine itself consists of a moving copy board which holds the drawing to be photographed and is adjustable in its angle of slant, an adjustable device for setting the position of the copy on the board, an electric motor to move the board forward and back, a camera for recording the axonometric projection, a funnel which moves vertically and restricts the camera's view to a narrow slot in the end, a 3000-watt light tube for illuminating the negative, a stand which supports the rolling fulcrum of the funnel, and a movable stand which supports the camera.

A negative of the drawing to be projected is placed on the glass surface of the copy board supported on two sides by the positioning device which can be rotated according to a scale on its outer perimeter. While the negative is in the horizontal position, the copy board is placed at the proper angle for the plane desired and the camera is moved backward or forward to produce the required scale. Since horizontal lines are projected in true proportion as long as the negative is in a horizontal position, the scale can be determined by measuring directly on the ground glass of the camera.

After the camera is placed and focused the negative is positioned on the copy board by rotating the position-adjusting device. As in the case of the copy board slant, the negative is positioned on the board at a predetermined angle from the horizontal depending on the plane in which the projection is being made. After the negative has been positioned, the camera is loaded with ordinary photographic printing paper and the machine is placed in operation.

(*Concluded on Page* 188)

MACHINE *Editorial* DESIGN

Current Trends Forecast After-the-War Developments

THOUGH no drastic switches in actual production yet have been made, the day is not far off when restrictions will be eased both on materials and on the production of certain needed classes of civilian goods.

Donald M. Nelson has had a hard time convincing the Army and Navy of the vital need for gradual reconversion of industry, gaining his point only when assurance could be given that any reconversion that might take place would not interfere directly with war production. In other words, when materials, labor and plant facilities can be released.

In order to fulfill its responsibility in the reconversion to peace as adequately as, it is hoped, it played its part in the conversion to war, MACHINE DESIGN has seized every opportunity to cover the numerous trends toward adoption of new materials, new parts and new methods that will be utilized in machines for the postwar market. Actual details of design of such machines cannot, of course, yet be divulged, but the expectation is that this will be feasible in the not-too-distant future.

Though the articles published during the more recent years of the war have not, in many cases, been presented strictly from the "postwar" angle, the information which they disclose will be recognized by alert engineers as forming the basis upon which many a postwar design—or entirely new development—is being and will be created.

Dividends will accrue to those designers able to keep track of significant trends and to apply such knowledge in their work. This is particularly the case in the relatively uncertain days that lie ahead.

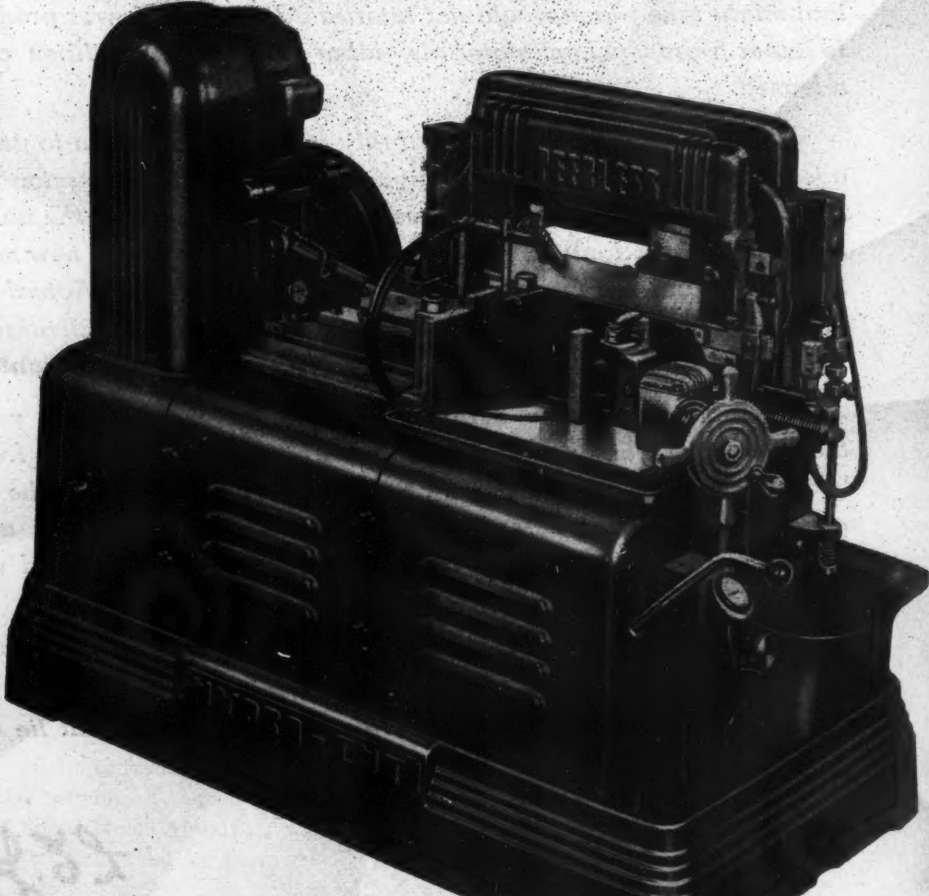
L. E. Jermy

Outstanding Designs

Hydraulic Hack Saw Machine

A compact, self-contained hydraulic feed unit is incorporated in this hack saw machine made by Peerless Machine Co. Blade feed pressure, supplied by a gear-type hydraulic pump, is controlled by a finger-tip pressure-control valve with gage attached. The feed and lift control piston is actuated by a cam which alternately directs the flow of hydraulic fluid for feed and lift cycles. Amount of lift of the saw blade is controlled by a displacement piston that permits specific amount of oil to exhaust from feed cylinders.

At completion of the cutting cycle, a trip valve actuates a start and stop piston which returns the saw frame to position for the next cutting cycle and automatically stops the machine. Crosshead is of cast steel with all bearing surfaces faced with hardened and ground inserts. The crosshead slide is of hardened and ground steel.

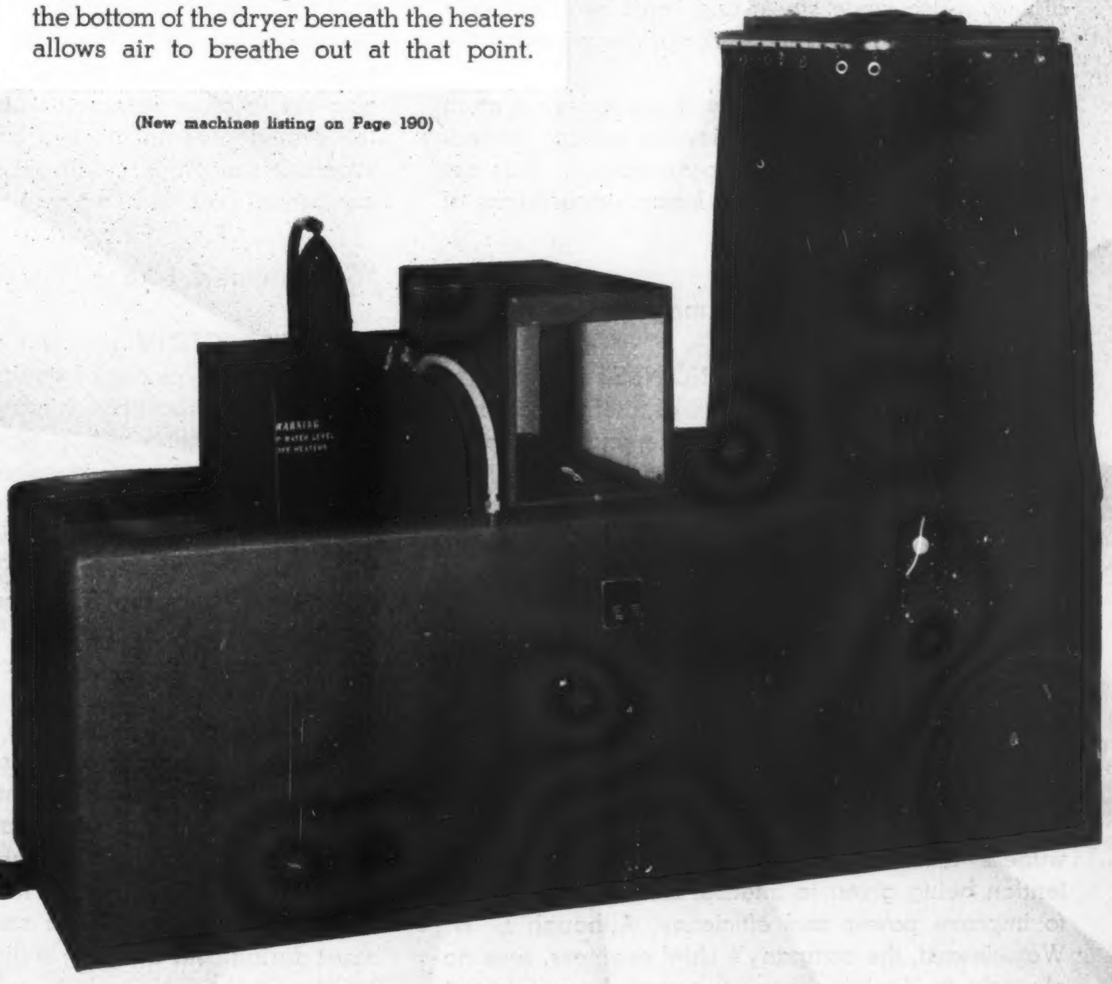


Fluorescent Inspection Machine

Over the penetrant tank at the left in this "Zyglo" inspection machine manufactured by Magnaflux Corp. is a removable half grill in which parts are placed for drainage. Immediately to the right of the penetrant tank is a rinse or wash station employing a spray nozzle and hand-operated valve connected to a standard water line. A black light is installed over the wash station. To the rear of the wash station is a tank of still water which is heated electrically.

There is no door over the front opening of the recirculating hot air dryer, the heated air being confined by a system of baffles. The design consists of strip electrical heaters in the lower front section arranged in a pattern to form their own baffling. In the rear lower section in a separate housing is a fan which draws in a small amount of fresh air continuously through the fan motor to keep it cool. A hole in the bottom of the dryer beneath the heaters allows air to breathe out at that point.

(New machines listing on Page 190)



Design Roundup

Current Thoughts on Hardenability

HARDENABILITY TESTS come close to being a complete specification for steel, but in practice they are inadequate as a single specification. Not enough properties are identified with hardenability to afford uniform manufacturing methods. For instance, manganese may be increased in an alloy to give hardenability of chrome vanadium steels but such steels do not have the same machinability nor same heat treatment as other alloys having the same hardenability. Also some steels and some sections have increased propensity for cracking, complicating the control of heat treatment.

One company has done considerable development on controlling hardening to produce surface stresses from tensile through zero to compression. This development promises to have future applications of significance.

Improving Engine Performance

ASIDE FROM AIRCRAFT ENGINES, the war created a need for large numbers of internal combustion engines for stationary, marine and transportation uses in which maximum power could be developed with minimum weight. This has been accomplished through utilization of the supercharger, power in some cases having been increased 20 to 30 per cent or considerably more. Convinced that the tendency toward lighter weight engines will stimulate even wider use of superchargers, Borg-Warner Corp., Chicago, some time ago acquired the McCulloch Engineering Corp., Milwaukee, a pioneer in this field, and just recently changed its name to B-W Superchargers Inc. This company is confident that the next few years will witness considerable design and development attention being given to internal combustion engines to improve power and efficiency. Although E. W. Wasielewski, the company's chief engineer, sees no obstacle to doubling engine power, he points out there are some limitations on the amount of power increase by supercharging. This particularly, is the case with the Otto-cycle or gasoline engines, where manifold temperatures and pressures must be kept within limits except as higher octane fuels permit operation at higher temperatures. Another factor is strength of the mechanical parts. However, with constantly improving engine design, materials and fuels, he is certain the supercharger will become increasingly important as manifold pressures rise.

To Control Large Power

DESIGN ENGINEERS may well reckon on the growing importance of the jet-propulsion engine. For, only recently Reginald E. Gillmor, president, Sperry Gyroscope Co., declared that an inspection of this type of engine, which is still a closely guarded military secret, showed that it was "remarkable in its simplicity and in the ease and flexibility of its control of very large power". These engines have also led to other research which will contribute to the evolution of an efficient gas turbine and other propulsion machinery with possibilities for decreasing capital cost and improving operating economy.

"Streamlined Like a Varga Girl"

FARM TRACTORS generally are regarded as relatively standard pieces of equipment, which in normal times undergo little change in design from year to year. According to L. B. Sperry, manager of engineering, farm tractor division, International Harvester Co., however, the postwar tractor may be a designers' paradise. Among features desired and which may be attempted, Mr. Sperry mentions: Engine starting by radio-activation, thus solving starting problems; torque converter transmission with automatic control of vanes and stator blades so that full efficiency always is secured; gyroscopic control of center of gravity to eliminate overturning hazards; electric eye steering so that obstructions in field operations are by-passed automatically; radio-activated, quickly attachable farm implement control without driver leaving the seat; automatic coupling device for drawbar-operated machinery; deluxe seat with collapsible cab which can be heated by exhaust during cold weather; styling as smooth as face powder and streamlined like a Varga girl; and two-way radio. All sounds fantastic now, of course, but who knows but that the future may see much of this fulfilled?

Direct Method Facilitates Helical Spring Design

By R. G. Minarik

Professor of Mechanical Engineering
Syracuse University

ALTHOUGH procedures, tables and charts are available for use in the design of helical springs, it frequently is found that they are inadequate for obtaining answers to many spring problems without much cut-and-try calculation. The design procedure presented in this data sheet is an effective and proved routine for the rapid solution of such problems. It is inherently time-saving and accurate, can be handled with confidence by relatively inexperienced personnel and, because of its pictorial nature and flexibility, is adaptable to problems involving a wide variety of imposed design conditions. The value of this routine method is further enhanced by the fact that it automatically involves steps which eliminate the ever-present chance of the designer's overlooking such factors as the correction of stress for coil curvature and the compensation of stress and deflection values for permissible stresses and shear moduli which may be different than those that have been used in the compilation of handbook tables and charts.

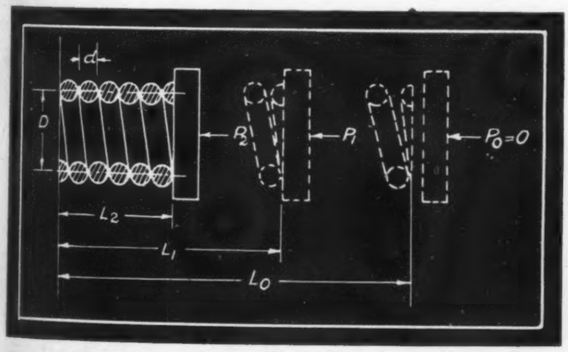


Fig. 1—Notation used in design of compression springs

Basic equations used for establishing the chart and formulas are the generally accepted equations for helical springs of normal proportions:

$$S=\frac{8PDK}{\pi d^3}$$

$$y=\frac{\pi n S D^2}{KdG}$$

where

S = Maximum induced shear stress, pounds per square inch
y = Total deflection, inches
P = Maximum load (inducing S) pounds

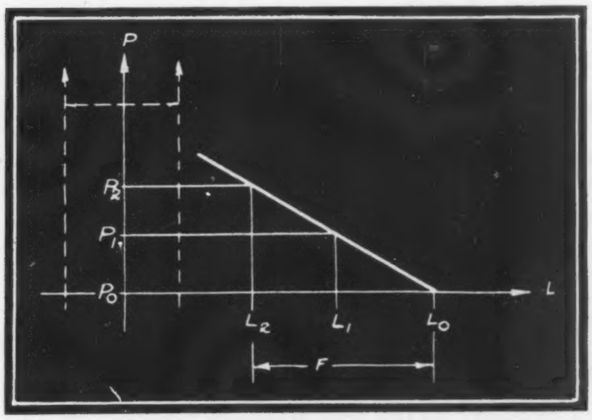
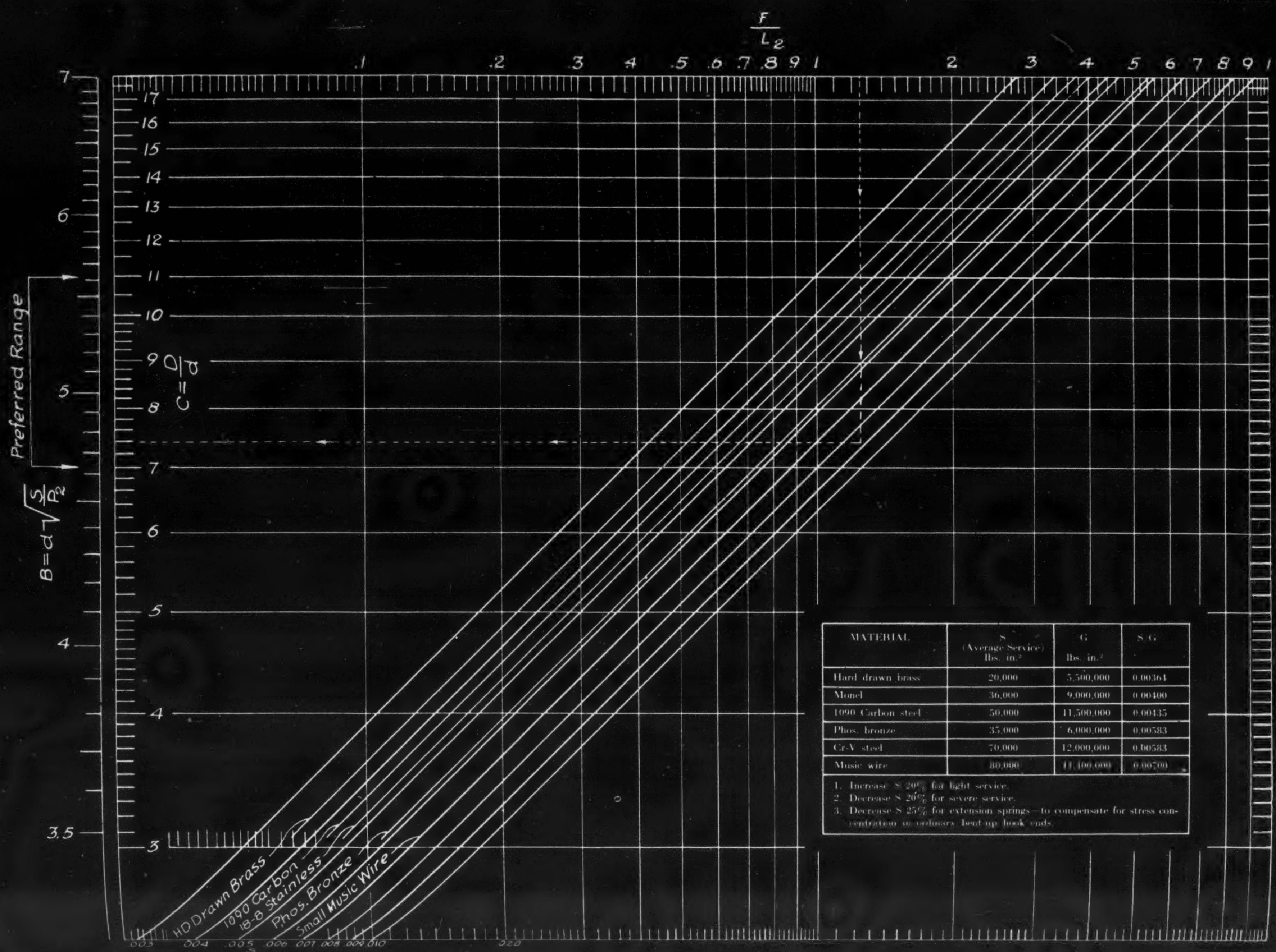


Fig. 2—Load-deflection diagram for compression spring

D = Mean (pitch) diameter of coil, inches
d = Wire diameter, inches
C = D/d
K = Curvature correction (Wahl factor)

$$=\frac{4C-1}{4C-4}+\frac{.615}{C}$$

n = Number of active turns
G = Shear modulus, pounds per square inch.



MATERIAL	S (Average Service) lbs. in.²	G lbs. in.²	S/G
Hard drawn brass	20,000	5,500,000	0.00364
Monel	36,000	9,000,000	0.00400
1090 Carbon steel	50,000	11,500,000	0.00435
Phos. bronze	35,000	6,000,000	0.00583
Cr-V steel	70,000	12,000,000	0.00583
Music wire	80,000	11,400,000	0.00700

1. Increase S 20% for light service.
2. Decrease S 20% for severe service.
3. Decrease S 25% for extension springs—to compensate for stress concentration in ordinary bent-up hook ends.

Compression Springs

In the design of compression springs, the notation shown in *Fig.* 1 is employed. The illustration shows the ends of the spring ground square, hence the active lengths and actual lengths are equal. Referring to *Fig.* 1:

L_0 = Free length of spring

L_1 = Length of spring corresponding to an imposed load P_1 $(0<P_1<P_2)$

L_2 = Fully compressed length of spring (adjacent coils just touching).

All lengths defined in the foregoing are "active" lengths (center line to center line of ends of the active coils) and are in inches.

P_1 = Load, pounds, required to compress spring to length L_1

P_2 = Load required to compress spring fully to length L_2 (this load induces maximum stress)

R = Spring rate, pounds per inch.

Procedure

Following is the procedure in designing a compression spring:

1. Draw the load-deflection line of the spring accurately to any convenient scales for P and L, as shown in *Fig.* 2.

Difference between the free and the fully compressed lengths, $L_2 - L_0$, is denoted by F. The load-deflection line can be determined from many combinations of data—usually obtainable directly from the design layout and performance requirements. In a few cases, when the performance requirements are not strictly defined, the designer must arbitrarily provide numerical values which will enable him to draw the load-deflection line.

2. Compute the value of F/L_2.

For conservatively proportioned springs of carbon steel, the preferred value of F/L_2 lies between .5 and 1.5, with 1.0 as an average. Therefore, for average proportions, F should approximately equal L_2 in *Fig.* 2. If the value of F/L_2 does not come in the preferred range, and there are no physical restrictions upon L_2 and L_0, the value of F/L_2 can be brought into the desired range by simply moving the P axis (movement shown dotted in *Fig.* 2) an appropriate amount to the right or left as may be required to alter L_2. If there are strict limitations on L_2 and L_0, and consequently the P axis cannot be shifted so as to give a preferred value of F/L_2, the calculation may still proceed in the normal manner here outlined, although the resulting proportions of the spring probably will not be ideal.

3. Locate the value of F/L_2, just determined, on the upper horizontal scale of *Fig.* 3. Project vertically downward to the 45-degree diagonal line corresponding to the selected material (or value of S/G) and then horizontally to the left to obtain values of C and B from the vertical scales.

Fig. 3—Opposite page—Chart for solving problems in the design of helical springs

C is the ratio D/d (spring index) and should be within the preferred range shown in *Fig.* 3 for best proportions. If a preferred value of C does not result from a given value of F/L_2 (or an altered value as obtained from the procedure discussed in Step 2), a value of C as low as 3 or as high as 18 is allowable—but not particularly desirable.

B is a value used in computing the wire diameter and related dimensions, and is equal to $d\sqrt{(S/P_2)}$.

If a material other than those shown in *Fig.* 3 is involved, or a different allowable maximum shear stress (S) is desired—as would be the case for severe service—a new 45-degree diagonal, with an intercept equal to the appropriate value of S/G on the horizontal scale at the bottom of the chart, should be determined. Visual interpolation usually is more convenient than the actual drawing in of the new diagonal.

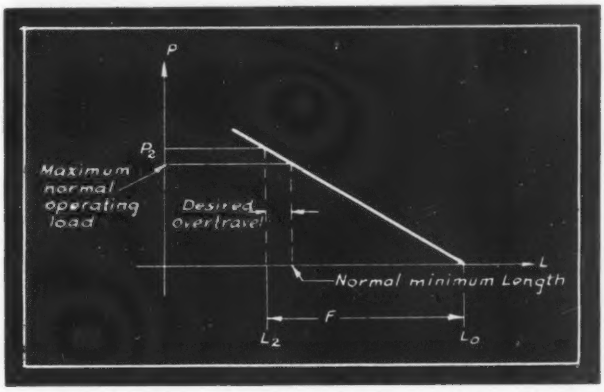


Fig. 4—Load-deflection diagram for compression spring with coil clearance at maximum compression

4. Calculate wire diameter $= d = B\sqrt{(P_2/S)}$, inches, (where S is the same as used in evaluating S/G in Step 3).

5. Mean (pitch) diameter $= D = C \times d$, inches.

6. Outside diameter (O.D.) $= D + d$, inches.

7. Inside diameter (I.D.) $= D - d$, inches.

8. Spring rate $= R = P_2/F$, pounds per inch.

9. Number of active turns $= n = L_2/d$.

10. Actual number of turns: To computed value of active turns (n), add:

0 for plain or ground ends

2 for squared or squared and ground ends.

11. Actual lengths: To lengths (L) from load-deflection diagram, such as *Fig.* 2, add:

d for plain ends
0 for ground ends
$3d$ for squared ends
$2d$ for squared and ground ends.

Design with Coil Clearance

In many cases coil clearance is desirable at the minimum operating length, particularly when better spring action is required and when the results of overtravel on a fully closed spring would provoke failure of the mechanism of which the spring is a component part.

Design of a compression spring having coil clearance at its minimum operating length proceeds in exactly the same manner as previously outlined with but one exception—a change in Step 1. The initial step for this case is designated 1*a* and is as follows:

1*a*. Draw the load-deflection line as shown in *Fig.* 4. It will be noted that here P_2 is determined by both the desired normal maximum operating load and the desired overtravel (often taken as 10 per cent of F). Remaining steps are exactly as before.

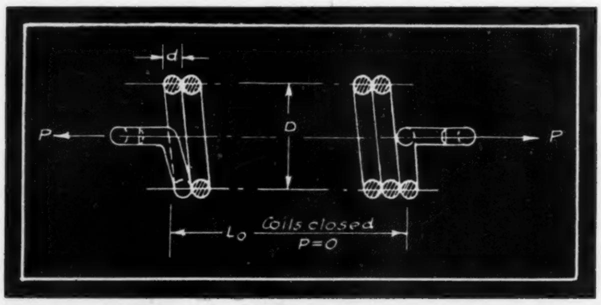


Fig. 5—Notation used in design of extension springs

Extension Springs

Design of helical extension springs is carried out in a manner similar to that employed for compression springs. The principal variation lies in the redefinition of some symbols in accordance with *Figs.* 5 and 6.

1. Draw the load-deflection line as shown in *Fig.* 6.

2. Evaluate F/L_0 and S/G. (Refer to the table on *Fig.* 3 regarding values of S to be employed.) Adjust F/L_0 for preferred range of C if desired.

3. Enter *Fig.* 3 with the value of F/L_0 on the F/L_2 scale; and find values of C and B.

4. Wire diameter $= d = B\sqrt{(P_2/S)}$, inches, (where S is the same as used in evaluating S/G in Step 2)

5. Mean (pitch) diameter $= D = C \times d$, inches.

6. Number of active turns $= n = L_0/d$ (coils closed at zero load).

7. Outside diameter (O.D.) $= D + d$, inches.

8. Inside diameter (I.D.) $= D - d$, inches.

9. Spring rate $= R = P_2/F$, pounds per inch.

10. Actual lengths and actual turns depend upon method used for attaching the spring.

EXAMPLE: It is required to find the dimensions of a compression spring for the following conditions: Ends squared and ground, spring rate $R = 100$ pounds per inch, total compression (fully open to fully closed) $= 2.5$ inches, fully compressed length $= 2$ inches minimum, (therefore take $L_2 = 2$ inches). Material is alloy steel with $S = 100{,}000$ pounds per square inch and $G = 11{,}500{,}000$ pounds per square inch.

SOLUTION: On *Fig.* 2 lay off $L_2 = 2$ inches and $F = 2.5$ inches, which makes $L_0 = 4.5$ inches. From L_0 draw the load-deflection line with a slope of 100 pounds per inch of deflection, giving $P_2 = 250$ pounds. With $F/L_2 = 2.5/2 = 1.25$, enter *Fig.* 3 at the top and drop a perpendicular to a point corresponding to $S/G = 100{,}000/11{,}500{,}000 = .0087$, interpolating between the 45-degree lines for $S/G = .008$ and $.009$. A horizontal line from this point intersects the C-scale at 7.4 and the B-scale at 4.75.

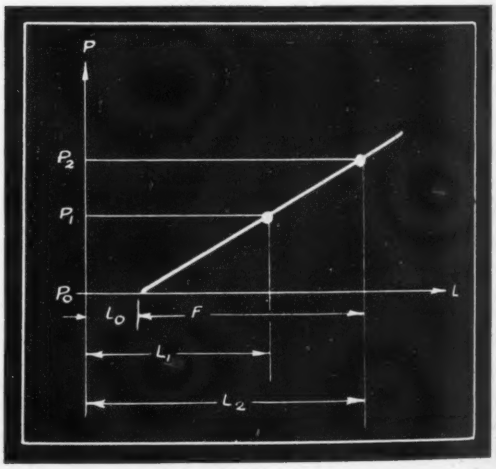


Fig. 6—Load-deflection, extension springs

Spring dimensions may therefore be calculated as follows: Wire diameter $d = B\sqrt{(P_2/S)} = 4.75 \times \sqrt{(250/100{,}000)} = .238$-inch; mean (pitch) diameter $= C \times d = 7.4 \times .238 = 1.76$ inches; number of turns $n = L_2/d = 2/.238 = 8.4$ turns, say 8½. Actual turns $= n + 2 = 8½ + 2 = 10½$ turns; actual free length $= L_0 + 2d = 4.975$, say 5 inches; actual closed length $= L_2 + 2d = 2.475$, say 2½ inches.

Materials Work Sheet

Filing Number 11.00

Brasses

ASTM No. B36-43T, Alloys 1 to 8
B134-42T, Alloys 1 to 8
B135-43T, Alloys 1 to 5

AVAILABLE IN:

(B36-43T) .... Sheet, strip and plate
(B134-42T .... Wire and rod
(B135-43T) .... Tubing

ANALYSES

B-36-43T

Alloy No.	Copper	Lead max.	Iron max.	Zinc*
1	94 to 95	.03	.05	..
2	89 to 91	.05	.05	..
3	84 to 86	.05	.05	..
4	79 to 81	.05	.05	..
5	70.5 to 73.5	.05	.05	..
6	68.5 to 71.5	.05	.05	..
7	66 to 69	.07	.05	..
8	64 to 67.5	.3	.05	..

B134-42T

Alloy No.	Copper	Lead max.	Iron max.	Zinc*
1	94 to 96	.05	.05	..
2	89 to 92	.05	.05	..
3	84 to 86	.05	.05	..
4	79 to 81	.05	.05	..
5	71 to 73	.05	.05	..
6	68.5 to 71.5	.07	.05	..
7	64 to 67.5	.10	.05	..
8	61 to 64	.10	.05	..

B135-43T

Alloy No.	Copper	Lead max.	Iron max.	Zinc*
1	83 to 86	.06	.05	..
2	68.5 to 71.5	.075	.06	..
3	65 to 68	.3 to .8	.07	..
4	65 to 68	1.25 to 2	.07	..
5	58 to 63	.8	.07	..

*Zinc constitutes remainder in all alloy analyses.

PROPERTIES

TENSILE STRENGTH

(psi)

	Alloys							
	1	2	3	4	5	6	7	8
SHEET AND STRIP (Spec. B36)								
¼ hard, min	37,000	40,000	44,000	48,000	49,000	49,000	49,000	49,000
max	47,000	50,000	54,000	58,000	59,000	59,000	59,000	59,000
½ hard, min	42,000	47,000	51,000	54,500	56,500	56,500	55,000	55,000
max	52,000	57,000	61,000	64,500	66,500	66,500	65,000	65,000
¾ hard, min	46,000	52,000	57,000	61,000	64,000	64,000	62,000	62,000
max	56,000	62,000	67,000	71,000	74,000	74,000	72,000	72,000
Hard, min	50,000	57,000	63,000	68,000	71,000	71,000	68,000	68,000
max	59,000	66,000	72,000	77,000	81,000	81,000	78,000	78,000
Extra hard, min	56,000	64,000	71,500	78,000	82,500	82,500	79,000	79,000
max	64,000	72,000	79,500	87,000	91,500	91,500	88,500	88,500
Spring, min	59,500	69,000	78,000	85,000	90,500	90,500	86,000	86,000
max	67,500	77,000	86,000	93,000	99,500	99,500	95,000	95,000
Extra spring, min	61,000	72,000	82,000	89,000	95,000	95,000	89,500	89,500
max	69,000	80,000	90,000	97,000	104,000	104,000	98,500	98,500
WIRE .02 TO .25-IN. DIAMETER (Spec. B134)								
⅛ hard, min	35,000	38,000	43,000	50,000	50,000	50,000	50,000	50,000
max	45,000	50,000	57,000	65,000	66,000	66,000	65,000	65,000
¼ hard, min	41,000	45,000	53,000	62,000	63,000	63,000	62,000	62,000
max	51,000	57,000	65,000	75,000	78,000	78,000	77,000	77,000
½ hard, min	49,000	56,000	66,000	78,000	80,000	80,000	79,000	79,000
max	58,000	67,000	77,000	90,000	95,000	95,000	94,000	94,000
¾ hard, min	57,000	64,000	76,000	90,000	93,000	93,000	92,000	92,000
max	64,000	74,000	86,000	101,000	108,000	108,000	107,000	107,000
Hard, min	61,000	70,000	83,000	100,000	103,000	103,000	102,000	102,000
max	68,000	79,000	92,000	110,000	118,000	118,000	117,000	117,000
Extra hard, min	66,000	78,000	94,000	112,000	117,000	117,000	115,000	115,000
max	73,000	86,000	102,000	121,000	131,000	131,000	129,000	129,000
Spring, min	72,000	84,000	100,000	120,000	126,000	126,000	123,000	123,000
DRAWN TUBING (Spec. B135)								
Light drawn, min	44,000	52,000	50,000	50,000	54,000			
max	60,000	70,000	68,000	68,000	72,000			
Hard drawn, min	60,000	70,000	68,000	68,000	72,000			

MACHINE DESIGN is pleased to acknowledge the collaboration of the following companies in this presentation: The American Brass Co.; The Baltimore Brass Co.; Chase Brass & Copper Co.; Revere Copper & Brass Inc.; The Riverside Metal Co.; Scovill Manufacturing Co.

Materials Work Sheet

ELONGATION IN 2 INCHES
(typical values, per cent)

	Alloys 1	2	3	4	5	6	7	8
SHEET AND STRIP .04. THK. (Spec. B36)								
¼ hard	25	25	25	30	43	43	43	43
½ hard	14	11	10	18	23	23	23	23
¾ hard	8	9	9	13	..	15	..	15
Hard	5	5	5	7	8	8	8	8
Extra hard	4	4	4	5	5	5	5	5
Spring	4	3	3	3	3	3	3	3
Extra spring	4	3	3	3	3	3	3	3
WIRE .08-IN. DIAM. (Spec. B134)								
⅛ hard	..	32	..	25	45	45	45	45
¼ hard	..	13	..	12	20	20	20	20
½ hard	..	6	..	8	..	15	15	15
¾ hard	..	5	..	..	..	10	..	10
Hard	..	4	..	5	..	8	8	8
Extra hard	..	3	..	4	4	4	4	4
Spring	..	3	..	3	3	3	3	3
DRAWN TUBING (Spec. B135)								
Light drawn	25	20	18	18	18			
Hard drawn	8	8	7	7	10			

SHEAR STRENGTH
(typical values, psi)

	Alloys 1	2	3	4	5	6	7	8
SHEET AND STRIP (Spec. B36)								
¼ hard	32,000	33,000	35,000	36,000	36,000	36,000	.....	36,000
½ hard	34,000	35,000	37,000	39,000	40,000	40,000	.....	40,000
¾ hard	35,000	36,000	40,000	41,000	.....	42,000	.....	42,000
Hard	37,000	38,000	42,000	43,000	44,000	44,000	.....	43,000
Extra hard	39,000	40,000	44,000	45,000	46,000	46,000	.....	45,000
Spring	40,000	42,000	46,000	48,000	48,000	48,000	.....	47,000
Extra spring	41,000	43,000	47,000	49,000	.....	49,000	.....	49,000
WIRE .02 TO .25-INCH DIAMETER (Spec. B134)								
⅛ hard	.....	33,000	35,000	37,000	.....	38,000	.....	.....
¼ hard	.....	34,000	38,000	42,000	.....	.....	42,000	.....
½ hard	.....	37,000	43,000	47,000	.....	.....	.....	.....
¾ hard	.....	40,000	45,000	.....	.....	.....	.....	.....
Hard	.....	42,000	48,000	53,000	.....	.....	55,000	.....
Extra hard	.....	43,000	52,000	.....	.....	.....	.....	.....
Spring	.....	45,000	54,000	60,000	.....	60,000	60,000	.....

PHYSICAL CONSTANTS

Alloys (ASTM Spec's. and Commercial Names)	Melting Point (deg C)	Young's Modulus (10^6 psi)	Specific Grav.	Weight (lb. per cu in.	Thermal Conductivity (Btu/sq ft /ft/hr/deg F (68° F)	Thermal Coef. of Linear Expansion per deg F 77-572° F	Electrical Resistivity (ohms/mil ft) at 20° C	Electrical Conductivity (% of IACS) at 20°C	Temp. Coef. of Electrical Resistance (/deg C) 20-200°C
B36, B134, Alloy 1 (Gilding Metal)	1065	16	8.86	.32	137	.0000101	18.5	56	.00231
B36, B134, Alloy 2 (Commercial Bronze)	1045	15	8.8	.318	109	.0000101	23.5	44	.00186
B36, B134, Alloy 3; B135, Alloy 1 (Red Brass)	1025	15	8.75	.316	92	.0000104	28	37	.0016
B36, B134, Alloy 4 (Low Brass)	1000	15	8.66	.313	81	.0000106	32	32	.00154
B36, B134, Alloy 5 (72/28 Brass)	965	15	8.55	.309	71	.0000110	32	28	.00148
B36, B134, Alloy 6; B135, Alloy 2 (70/30 Brass)	955	15	8.53	.308	70	.0000111	37	28	.00148
B36, Alloy 7 (2 & 1 Brass)	940	15	8.47	.306	67	.0000112	40	26	.00154
B36, Alloy 8; B134, Alloy 7 (High Brass rolled flats)	930	15	8.47	.306	67	.0000112	40	26	.00154
B134, Alloy 8 (High Brass wire)	920	15	8.44	.305	67	.0000114	40	26	.00154
B135, Alloy 5 (Muntz Metal)	905	14	8.39	.303	71	.0000116	37	28	.00198
B135, Alloy 3 (Low-Leaded Brass tube)	940	15	8.5	.307	67	.0000112	40	26	.....
B135, Alloy 4 (High-Leaded Brass tube)	930	15	8.53	.308	67	.0000113	40	26	.....

YIELD STRENGTH

(typical values—.5% elongation under load, psi)

	Alloys 1	2	3	4	5	6	7	8
SHEET AND STRIP (Spec. B36)								
¼ hard	32,000	34,000	40,000	40,000	40,000	40,000	40,000	40,000
½ hard	40,000	45,000	50,000	50,000	52,000	52,000	51,000	50,000
¾ hard	46,000	50,000	54,000	55,000	.....	58,000	.....	58,000
Hard	50,000	54,000	58,000	58,000	64,000	64,000	62,000	60,000
Extra hard	55,000	58,000	61,000	61,000	66,000	66,000	64,000	62,000
Spring	58,000	63,000	62,000	62,000	66,000	66,000	64,000	62,000
Extra spring	60,000	68,000	68,000	68,000	66,000	66,000	.....	62,000
DRAWN TUBING (Spec. B135)								
Light drawn	42,000	50,000	50,000	50,000	45,000			
Hard drawn	58,000	64,000	60,000	60,000	55,000			

ROCKWELL HARDNESS—B SCALE

	Alloys 1	2	3	4	5	6	7	8
SHEET AND STRIP (Spec. B36)†								
¼ hard, min	20	27	33	38	40	40	40	40
max	52	56	62	65	65	65	65	65
½ hard, min	40	50	56	59	60	60	57	57
max	60	66	71	73	77	77	74	74
¾ hard, min	50	59	66	69	72	72	70	70
max	64	71	76	79	82	82	80	80
Hard, min	57	65	72	76	79	79	76	76
max	67	75	80	84	86	86	84	84
Extra hard, min	64	72	78	83	85	85	83	83
max	72	79	85	89	91	91	89	89
Spring, min	68	76	82	87	89	89	87	87
max	75	81	87	92	93	93	92	92
Extra spring, min	69	78	84	88	91	91	88	88
max	76	83	89	93	95	95	93	93
DRAWN TUBING (Spec. B135)‡								
Light drawn, min	34	50	48	48	54			
max	70	78	75	75	82			
Hard drawn, min	70	80	78	78	82			

ROCKWELL HARDNESS—SUPERFICIAL 30-T SCALE

	Alloys 1	2	3	4	5	6	7	8
SHEET AND STRIP (Spec. B36)†								
¼ hard, min	29	34	38	42	43	43	43	43
max	51	54	58	60	60	60	60	60
½ hard, min	43	50	54	56	56	56	54	54
max	56	61	64	66	68	68	66	66
¾ hard, min	50	56	60	63	65	65	65	65
max	60	64	68	70	72	72	71	71
Hard, min	54	60	65	68	70	70	68	68
max	62	67	70	73	74	74	73	73
Extra hard, min	60	65	69	72	74	74	73	73
max	65	70	74	76	77	77	76	76
Spring, min	62	68	72	75	76	76	75	75
max	67	71	75	78	78	78	78	78
Extra spring, min	63	69	73	76	77	77	76	76
max	68	72	76	79	79	79	79	79
DRAWN TUBING (Spec. B135)‡								
Light drawn, min	38	52	50	50	55			
max	64	69	68	68	71			
Hard drawn, min	64	69	68	68	71			

† ASTM values for information only. They apply as follows: B scale applies to metal .02-inch thick and over. The 30-T scale applies to metal .012-inch thick and over.

‡ B scale values apply to tubes .02-inch and over in wall thickness. The 30-T scale values apply to tubes .012-inch and over in wall thickness.

CHARACTERISTICS

Composed wholly or mainly of copper and zinc, all brasses are known as copper or copper-base alloys. Often small percentages of such metals as lead, tin and aluminum are added to improve certain properties for specific applications. It will be seen from the chemical analyses tables that the difference between the various brass alloys is primarily in the ratio of copper to zinc. In general, the corrosion resistance of all brasses increases in proportion to their copper content while their strength, hardness and wearing qualities are improved proportionate to the zinc content. While all of these alloys can be cold worked, those with about 70 per cent copper content have the best physical properties when used for drawing or spinning. Extensive hot working is best effected on alloys with copper contents up to 63 per cent and over 84 per cent.

Alloys Nos. 1 and 2 (B36, B134): Because of their color, which resembles that of true bronze, these alloys commonly are known as "commercial bronze". Since they are less expensive than true bronze they often are substituted for it. Hot-working properties are excellent, cold-working properties are good, and welding can be effected by several of the commonly known methods. They provide an excellent base for enamel finishes.

Alloy No. 3 (B36, B134); **Alloy 1** (B135): Known commer-

Materials Work Sheet

cially as red brass or rich low brass. Has excellent cold working properties; can be spun, drawn, stamped, cold forged and upset. Its hot-working properties are rated good and are much the same as those of copper in some operations. Like copper, lead content is held to a trace to avoid cracking during hot working. This alloy is preferred to copper for resisting the action of salt, brackish or corrosive inland waters. Consequently it is used extensively for pipe and tubing in plumbing and heat exchangers. Differs from brasses of higher zinc content in that it is relatively immune to season cracking and dezincification.

Alloy No. 3 (B135): "Low-Leaded Brass." Excellent alloy for general-purpose brass tube, having good corrosion resistance, strength and surface finish. Often used in the manufacture of tubular products which require only a moderate amount of machining. Capacity for being expanded, bent, or otherwise cold worked is high.

Alloy No. 4 (B36, B134): "Low Brass." Hot-working operations are accomplished satisfactorily only when closely controlled and when the lead content is held to a trace. Corrosion resistance is good and resistance to season cracking and dezincification is slightly lower than in 85-15 red brass.

Alloy No. 4 (B135): Extensively used for screw-machine tubular products. Machinability is somewhat less than that of free-cutting brass.

Alloy No. 5 (B36, B134): "72/28 Brass." Has best combination of strength and ductility of all the copper-zinc alloys. Has excellent cold-working properties and if lead content is below .05 per cent, hot-working properties are fair. Good resistance to corrosion.

Alloy No. 6 (B36, B134): "70/30 Brass." Also known as Cartridge Brass. Has excellent cold-working properties, being used primarily for deep drawing. Hot-working properties are considered fair but depend on the lead content, the presence of .05 per cent resulting in cracking at hot-working temperatures. Generally, however, this alloy does not contain over .01 per cent lead and as a result can be hot worked satisfactorily.

Alloy No. 7 (B36): "2:1 Brass." Widely used for deep-drawn parts, consequently is furnished principally in the annealed condition.

Alloy No. 8 (B36, B134): "High Brass." Has been known also as Common High Brass, Drawing Brass and Yellow Brass. Cold-working properties are excellent, thus it is used extensively for such fabrication methods as deep drawing, forming, spinning, etc. Has poor hot-working properties.

APPLICATIONS

All of the following applications pertain to alloys covered by ASTM Specifications B36 and B134 only.

Alloy No. 1: Fuse caps, primers, etc. Makes exceptionally good base for articles to be plated, highly polished or coated with vitreous enamel.

Alloy No. 2: Escutcheons, forgings, grillwork, hardware, kick plates, line clamps, primer caps, projectile rotating bands, rivets, screen wire, screws, screw shells, etc.

Alloy No. 3: Condenser tubes, electrical conduit, eyelets, fasteners, flexible hose, hardware, heat exchanger tubes, pickling crates, plumbing pipe, radiator cores, screw and socket shells, trim, etc.

Alloy No. 4: Battery caps, bellows, clock dials, drawn and formed parts, flexible hose, trim, etc.

Alloy No. 6: Cartridge cases, eyelets, rivets, springs, tubes, drawn and spun parts, etc.

Alloy No. 7: Bead chain, eyelets, fasteners, screw and socket shells, springs, deep-drawn parts, etc.

Alloy No. 8: Grillwork, gromets, screws, rivets, reflectors, springs, chain, eyelets, fasteners, hardware, kick plates, pins, push plates, radiator tanks and cores, screw and socket shells, blanked, spun and drawn parts, etc.

FABRICATION

MACHINABILITY:

All the alloys of Specifications B36 and B134 are machinable, but their machinability rating is considerably below that of free-cutting brass which owes its excellent machining characteristics to the presence of approximately 3 per cent lead in the alloy. Assuming free-cutting brass to have a machinability rating of 100, it is convenient to assign a rating of 20 to Alloys 1 and 2 of Specifications B36 and B134 and a rating of 30 to Alloys 3 to 8. The lead-bearing alloys 3 and 4 of Specification B135 have ratings of 60 and 80 respectively. These are not rigid classifications because machinability cannot be defined precisely and an alloy may vary in its behavior according to the circumstances under which it is handled. In general, machinability of these alloys improves as the copper content decreases. All of them produce long and tough turnings or chips.

DRILLING:

For drilling Alloys 3 to 8 (Spec's. B36, B134) it often is found advisable to grind the rake of a standard twist drill to zero degrees and flatten the cutting edge approximately 6 to 8% of the drill diameter. Standard twist drills are used satisfactorily without alteration for Alloys 1 and 2 (Spec's. B36, B134), although the so-called high-spiral drills having a helix angle of about 40 degrees are particularly suitable, especially for deep drilling. Although these materials sometimes are drilled dry, lubrication is recommended.

When using high-speed steel drills with lubricant, speeds of 50 to 125 feet per minute are suitable for Alloys 1 and 2, and 75 to 250 feet per minute for Alloys 3 to 8. Feeds normally range from approximately .003 to .02-inch per revolution depending on drill size.

REAMING:

When specifying the size of drilled hole to precede reaming, leave sufficient stock in the hole, especially where wall thickness is small in relation to the hole diameter. Such precaution will insure against burnishing action which results in undersize holes and excessively rapid wear of the reamer. Straight-fluted reamers have a tendency to chatter on some types of work. Standard spiral-fluted reamers with 7 to 12-degree helix angle work satisfactorily, while left-hand spiral, right-hand cutting types give excellent results.

TAPPING:

Except in the comparatively few cases where an extremely close thread fit is required, it is advisable to specify a tap-drill size which will result in a 75% depth of thread. 100% threads are only 5% stronger than 75% threads, but require more than twice the power to tap and, in addition, present problems of chip ejection and correct tap design for the particular metal. An ample supply of cutting fluid is used for all tapping operations.

HOT AND COLD WORKING:

Hot workability of these alloys is governed largely by the amount of lead present in them as an impurity. Lead in amounts greater than approximately .02 per cent leads to hot-shortness and seriously restricts the permissible range of tem-

perature for hot working. The alloys of higher copper content, such as Alloys 1, 2 and 3 of Specification B36 are more plastic at elevated temperatures than the others.

HOT AND COLD-WORKING PROPERTIES

Alloy (ASTM Spec. and Commercial Name)	Cold Working	Hot Working	Hot-Working Temperature (deg F)
B36, B134, Alloy 1 (Gilding Metal)	Excellent	Good	1400-1600
B36, B134, Alloy 2 (Commercial Bronze) . .	Excellent	Good	1400-1600
B36, B134, Alloy 3; B135, Alloy 1 (Red Brass)	Excellent	Good	1450-1650
B36, B134, Alloy 4 (Low Brass)	Excellent	Fair	1500-1650
B36, B134, Alloy 5 (72/28 Brass)	Excellent	Fair	1350-1550
B36, B134, Alloy 6; B135, Alloy 2 (70/30 Brass)	Excellent	Fair	1350-1550
B36, Alloy 7 (2 & 1 Brass)	Excellent	Poor	. . . .
B36, Alloy 8; B134, Alloy 7 (High Brass rolled flats)	Excellent	Poor	. . . .
B134, Alloy 8 (High Brass wire)	Good	. . . .	. . . .
B135, Alloy 5 (Muntz Metal)	Fair	Excellent	1150-1450
B135, Alloy 3 (Low-leaded Brass tube)	Excellent	Poor	. . . .
B135, Alloy 4 (High-leaded Brass tube)	Fair	Poor	. . . .

While most of the alloys have excellent cold-working properties, frequent annealing is advisable where heavy reductions are required in drawing, rolling, pressing, spinning, etc. Best of these alloys for deep drawing is 70/30 brass.

SUITABILITY FOR WELDING AND BRAZING

Alloy (ASTM Spec. and Commercial Name)	Silver Alloy Brazing	Oxyacetylene Welding	Carbon Arc Welding	Resistance Welding (Spt. & Sm.)
B36, B134, Alloy 1 (Gilding Metal) . . .	Excellent	Fair	Good	Poor
B36, B134, Alloy 2 (Commerc'l Bronze)	Excellent	Good	Good	Poor
B36, B134, Alloy 3; B135, Alloy 1 (Red Brass)	Excellent	Good	Good	Poor
B36, B134, Alloy 4 (Low Brass)	Good	Good	Fair	Poor
B36, B134, Alloy 5 (72/28 Brass) . . .	Good	Good	Fair	Fair
B36, B134, Alloy 6; B135, Alloy 2 (70/30 Brass) . . .	Good	Good	Fair	Fair
B36, Alloy 7 (2 & 1 Brass) . . .	Good	Good	Fair	Fair
B36, Alloy 8; B134, Alloy 7 (High Brass rolled flats) . . .	Good	Good	Fair	Fair
B134, Alloy 8 (High Brass Wire)	Good	Good	Fair	Fair
B135, Alloy 5 (Muntz Metal)	Good	Good	Fair	Fair
B135, Alloy 3 (Low-leaded Brass tube) . . .	Good	Fair	Fair	Fair
B135, Alloy 4 (High-leaded Brass tube) . . .	Good	Fair	Fair	Poor

ANNEALING

When it is necessary to anneal these alloys during fabricating operations, it generally will be found that a temperature in the range from 800 to 1200 degrees Fahr. will produce the desired characteristics. The temperature actually used for any alloy will depend somewhat on the characteristics desired, the lower temperatures producing harder, stiffer and less ductile material than the higher temperatures. Grain size is the best criterion of the degree of anneal and accurately determines the properties in the annealed state. In general, the small grain sizes around .015-millimeter are used where an excellent surface and considerable stiffness are required. Increased grain size up to perhaps as high as .035-millimeter for high-copper alloys or .05 to .07-millimeter for yellow brasses would be required where the degree of working or the complexity of the shape is increased. In cold-worked brass, internal stresses that may cause season cracking frequently can be relieved by heat treatment at about 480-520 degrees Fahr. which is below the temperature needed to produce any softening effect. The time of such treatment may vary from a few minutes to several hours depending on the kind of article and amount of material being treated.

RESISTANCE TO CORROSION

These alloys have generally good corrosion resistance but there is considerable variation in resistance according to composition. Usually the higher the zinc content of the brass the better its resistance to sulphur compounds. Alloys containing about 80 per cent of copper or less sometimes are subject to dezincification, a form of corrosion in which the zinc is selectively removed from the alloy resulting in the formation of porous masses or layers of copper in the article undergoing corrosion. In plumbing work the use of red brass (Alloy 1, B135) has eliminated failures by dezincification entirely.

GALVANIC CORROSION

Like copper, these alloys are more noble than most commercial metals and alloys. They are cathodic to aluminum, iron, magnesium and zinc, and the latter metals and their alloys are corroded galvanically when coupled with the brasses in the presence of reasonably strong electrolyte. Severity of the attack upon the less noble materials is dependent on the relative anodic and cathodic areas, strength of electrolyte, temperature, and polarization effects.

MATERIAL DESIGNATIONS

ASTM Alloy No.	Federal	Army	Navy
SHEET AND STRIP (ASTM Spec. B36-43T)			
1		57-171-2	
2		57-171	
3		57-160	
4		57-160	
6	QQ-B-611a Comp. E	Federal 57-172-1C	
7	QQ-B-611a Comp. C	Federal	47B2 (INT)
WIRE (ASTM Spec. B134-42T)			
5	QQ-W-321 Grade C	Federal	Federal
7	QQ-W-321 Grades A & B	Federal	Federal
TUBE (ASTM Spec. B135-43T)			
1	WW-T-791 Grade 1	Federal	44T15b. Grade 1 44T16 (INT)
3	WW-T-791 Grades 2 & 3	Federal	44T15b Grade 2

Correction: Work Sheet on Aluminum Alloys 2S, 3S and 52S appearing in July issue carried Filing No. 17.01. Filing number should have been 9.01.

ASSETS to a BOOKCASE

Automatic Control Engineering

By Ed Sinclair Smith, research engineer, Eclipse-Pioneer division, Bendix Aviation Corp.; published by McGraw-Hill Book Co. Inc., New York; 367 pages, 5¼ by 8¼ inches, clothbound; available through MACHINE DESIGN, *$4 postpaid.*

Automatic control being an extremely important phase of machine design and mechanical processing, it is rather surprising to find there has been a dearth of technical books published on the subject. Certainly there has existed a need for such a volume. Yet in spite of this most of the data appearing prior to this present work was to be found mainly in technical journals, engineering society papers and company publications.

Suffice it to say that within the pages of this text the reader will find a wealth of information on the characteristics and operating principles of numerous types of control units such as valves, meters and regulators as well as of the systems in which these units are employed. Many phases of design and utilization are presented including quantitative analysis, aspects of control, underlying physics, mathematics of transients, and analytic methods for the solution of differential equations. Considerable pertinent tabularized matter is offered and, for those who wish to pursue the study of control to its ultimate, unusually complete reference lists are supplied.

□ □ □

Mechanical Springs

By A. M. Wahl, Westinghouse Electric & Mfg. Co.; published by Penton Publishing Co., Cleveland; 435 pages, 6 by 9 inches; clothbound; available through MACHINE DESIGN, *$6 postpaid.*

Here at last is a complete text on the design, selection and specification of springs. The need for this book has long been felt by the machine design profession and that need has been given considerable emphasis in recent years by the demands of modern machines for more and more precision in design and construction. There are but few machines which utilize no springs whatsoever and in many the proper functioning of their various units is dependent largely on the skill with which the actuating springs have been selected and designed.

Often the machine designer has a choice of two or more spring types which might be specified for a particular application, and deciding which type will serve best can be quite puzzling. Should his decision be based predominantly on "rule of thumb" methods, the spring might fail in actual service because such factors as fatigue, internal stressing, creep effects, surging, buckling, etc., have not been accorded due consideration. However, once armed with authentic facts concerning the characteristics of the various spring types, he can proceed on a sound, rational basis which is free of guess work.

This, then, is a book which takes the uncertainties out of the use of mechanical springs. Dr. Wahl has been most thorough in his searching analyses of just what goes on in all kinds of springs under all kinds of conditions. His data are based on research conducted not only by himself, but by many contemporary authorities in the spring field.

Dealing first with the more elementary considerations, the discussion proceeds to helical tension and compression springs, Belleville, flat and leaf, torsion, spiral, ring, volute and rubber types, giving complete coverage to all phases of their design, selection and specification. The many graphs and tables presented give the reader quick access to stress, deflection and loading values and abundant drawings and photographs are employed to qualify the discussion throughout.

□ □ □

Graphical Solutions

By Charles O. Mackey, professor of heat power engineering, Cornell university; second edition; published by John Wiley & Sons Inc., New York; 152 pages, 5⅜ by 8 5/16 inches, clothbound; available through MACHINE DESIGN, *$2.50 postpaid.*

Where the designer uses the same formulas from day to day in his work there are few things which will expedite calculations for him as much as graphical solutions. While it is true that one occasionally meets an engineer who claims that he does not need to use the slide rule, the advisability of such procedure is open to question. There are many types of sliding scales which can be developed for solving various kinds of engineering equations. All that is needed is an understanding of the principles underlying the proportioning of the scales. These principles are expounded thoroughly in this fine little text book.

Of course there are many forms of graphical solution other than sliding scales that can be used to considerable advantage in design. There are, for example, the network or intersection charts for representation of equations containing three or more variables, alignment charts, with which all engineers are familiar, and the various curves, periodic and nonperiodic, which help so greatly in dealing with empirical equations.

If you are not now acquainted with the procedures used in developing graphical solutions, this book will show you how.

JOHNSON BRONZE SLEEVE TYPE BEARINGS

Reconversion

The time for complete reconversion to civilian production is when Victory is ours. The time to start thinking about it is NOW. When that great day comes, manufacturers of every line of products will be looking for a dependable source of supply for sleeve bearings . . . one that can deliver to exacting specifications—without delay.

This you will find at Johnson Bronze. While all of our present production is for essential war needs . . . our reconversion problem is relatively simple. All we need is a new set of prints.

When manufacturers place their bearing requirements with Johnson Bronze they gain the advantage of more than thirty-five years of exclusive bearing experience—plus complete facilities for the manufacture of every known type. When you start thinking about your postwar product, call in a Johnson Engineer. Permit him to review your applications . . . to make recommendations based on facts, free from prejudice. There is one located as near as your telephone.

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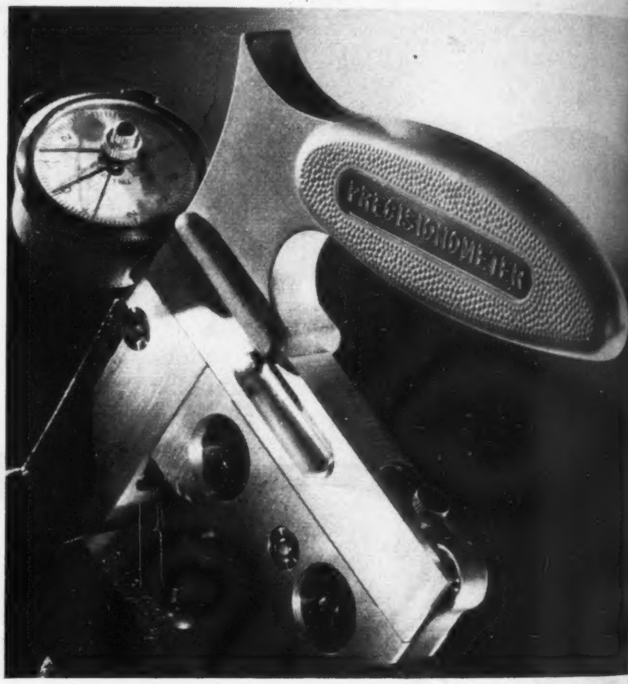
SLEEVE BEARING HEADQUARTERS
The MOST COMPLETE SLEEVE BEARING SERVICE in the WORLD
525 S. MILL STREET NEW CASTLE, PA.

Applications

of Engineering Parts, Materials and Processes

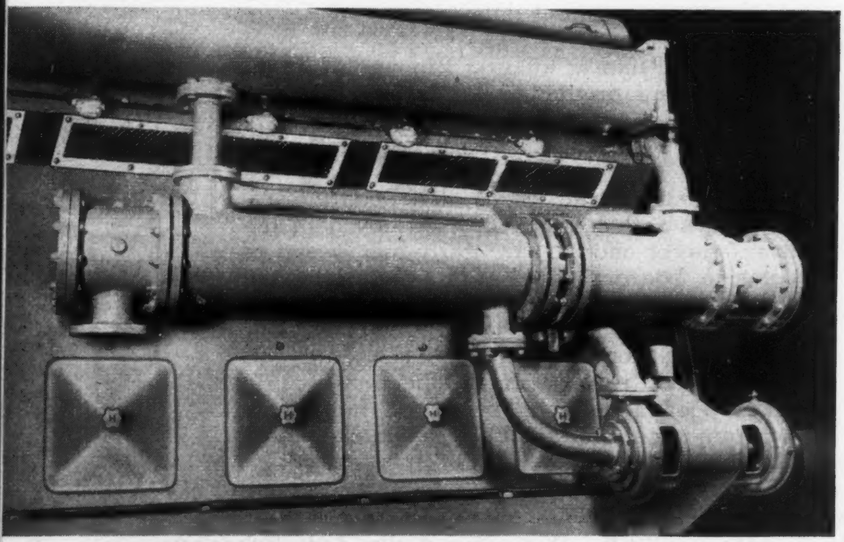
Plastic Handle Serves as Heat Insulator

AUTOMATIC compensation for temperature variations is combined with the employment of a low thermal-conductivity molded Tenite handle—preventing transmission of heat from the operator's hand—to achieve optimum precision in measurements made with the portable indicating comparator shown at right. As a further safeguard to accuracy the plastic handle is so designed that the operator's thumb will not touch the indicator. Injection-molded, it is made in two parts to provide a hollow handle of correct weight and balance.



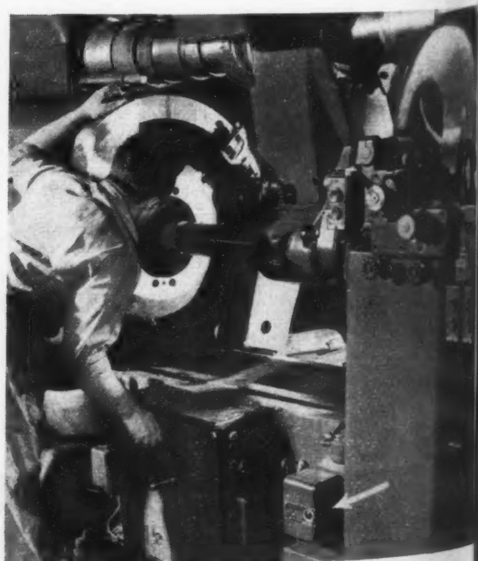
Diesel Employs Integral Heat Exchanger

INTEGRAL parts of Hendy Diesels are the tube-type cooling-water heat exchangers with fresh and raw water pumps and complete piping, shown in the photograph below. As a result, corrosion within the cylinder jackets and heads due to use of raw or salt water is eliminated and more efficient operating temperatures are maintained.



Times Grinding Period

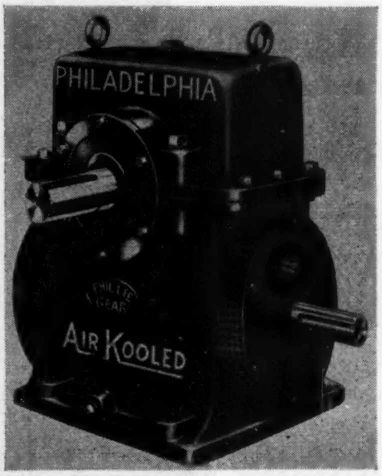
ELECTRONIC time-delay relay, indicated by arrow in illustration of grinder at right, times the period for grinding concentric bores in the stators of Tri-Clad motors. This G. E. timer is preset so that after a certain period, depending on the amount to be ground and other factors it automatically stops the grinding operation. Heart of the unit is a small electronic tube of the type used in home radio receivers. Because it has no continuously moving parts the timer is particularly useful for often-repeated, short-time cycles.



New PARTS AND MATERIALS

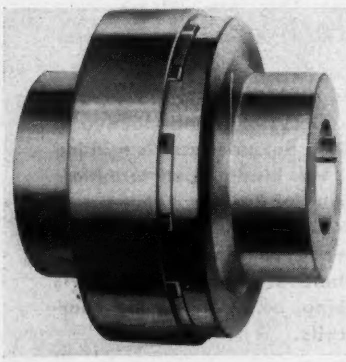
Aircooled Speed Reducer

OFFERING INCREASED horsepower ratings, the new AirKooled worm gear speed reducers introduced by Philadelphia Gear Works, Erie avenue and G street, Philadelphia 34, have worm speeds of 580 to 1750 revolutions per minute. In this type of aircooled unit heat is dis-



sipated by drawing cooling air through the double-walled housing with a suction fan mounted on the worm shaft. The reducers offer a saving in space and approximately a 40 per cent reduction in weight as compared with the standard unit of the same capacity, the latter figure being based on a worm speed of 1750 revolutions per minute at which the rating advantage is greatest.

Coupling Offers Protection



PROVIDING PROTECTION to operators, the new L-R Type "C" shrouded flexible coupling of Lovejoy Flexible Coupling Co., 5009 West Lake street, Chicago 44, is designed with a heavy steel collar holding the load cushions in place. An extension of this collar encircles the coupling and safeguards the material and fingers from the heads of the bolts that secure the load cushion retainer. Individual free-floating load cushions between the rugged jaws a held in position between an inside steel sleeve and th removable steel collar. Types are available for dire attachment to flywheel, also drum types and others f rapid disassembly without disturbing drive or drive equipment. Cushions of various materials suitable f the particular service on which the coupling is used ma be employed for correcting misalignment, protection fro shock and vibration, and other transmission difficultie In operation half the cushions are idlers (except on rever ing load). The noiseless coupling is designed for heav duty services from 4.6 to 806 horsepower at 100 revolu tions per minute.

Electronic Voltage Regulator

FOR USE WITH G-E resistance welding controls th include the phase-shift method of heat control, Gener Electric Co. offers its new electronic control. This ne voltage regulator is designed to regulate welding curre automatically so that it is held constant regardless o line-voltage variations of as much as plus 10 or min 20 per cent. Known as the CR7503-D157, it consists of resistor circuit connected to the power line supplying th welding machine, and an electronic circuit. When th regulator is in use, the electronic circuit functions not onl to hold the average voltage of the resistor current con

stant, but also changes, electronically, the phase-contr voltages of the main welding control panel. Thus if lin voltage drops, the regulator advances the phase-contr voltages, automatically holding the welding current con stant within close limits. Housed in a steel enclosure the unit can be used on welding circuits whose pow factor is from 20 to 70 per cent, and can be operat from 230/460/575 volts on a 50/60-cycle power suppl The regulator operates whether the welding current

VIM leather packings help control the muscles in these strong arms

Milwaukee comes clean! To collect rubbish, the city's Department of Public Works uses trucks equipped with special hoists built by The Heil Co. By the touch of a hand, telescopic cylinders raise the truck body and the rubbish slides out to its final resting place.

Sealing the hydraulic mechanism that provides the lift are VIM Leather Packings —capable of resisting pressures up to 100,000 PSI, and providing long-time service and lowest maintenance cost.

No matter what your design involves in hydraulics, Houghton's VIM Leathers, properly installed, will last longer. They are impregnated to be impervious to oil, air or gases. They're engineered correctly—a "plus" feature that appeals to designers.

To learn more about these packings and the design service our engineers render, call The Houghton Man or write—

E. F. HOUGHTON & CO.

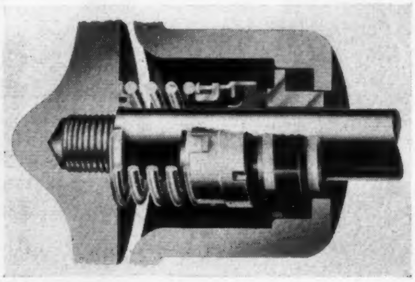
303 W. LEHIGH AVE. PHILADELPHIA 33, PA.

HOUGHTON'S Engineered VIM Leather Packings

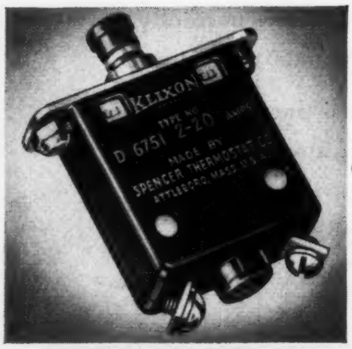
flowing or not, enabling it to compensate for sudden voltage drops in less than three cycles with approximately 75 per cent of the compensation taking place during the first cycle.

Bellows Type Shaft Seals

ELIMINATING stuffing-box leakage in centrifugal and rotary pumps, refrigeration compressors, speed reducers, agitators, gearboxes and other rotary-shaft equipment, the new bellows type shaft seal developed by The Crane Packing Co., 1825 Cuyler avenue, Chicago 13, has been tested on many exacting services. It operates at speeds as high as 5000 revolutions per minute, pressures to 300 pounds per square inch, temperatures to 240 degrees Fahr., and on shafts as large as 5⅝-inch diameter. The seal consists of three parts: A flexible synthetic rubber bellows which grips and seals along the shaft; a stationary seat, held in a synthetic rubber sealing ring; and a sealing washer which turns with the shaft and is held against the stationary seat by spring pressure. Contacting faces of the washer and seat are lapped to form a leakproof seal. The bellows construction permits the washer to remain in contact with the seat as the sealing faces wear. Correct working pressure is maintained as the flexible bellows offers no resistance to the spring.



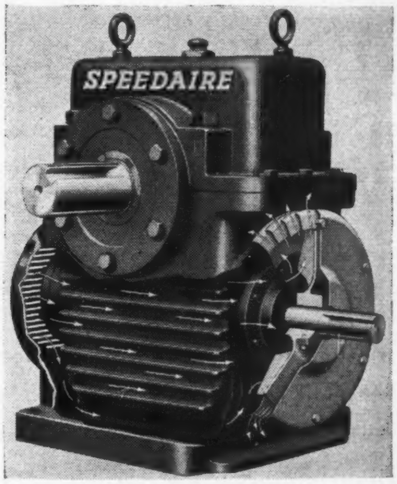
"Push-Pull" Circuit Breaker



CONFORMING to Navy specifications M-538 and NAF-1213, the new aircraft circuit breakers introduced by Spencer Thermostat Co., Attleboro, Mass., offer a "push-pull" feature. The pull permits the opening of the breaker manual for maintenance of the circuit without de-energizing the entire power system. To place the breaker in this position requires deliberate action which retains the desirable feature of being safe against accidental tripping when operating adjacent switches. The "push-pull" button travels outwardly approximately ¼-inch upon tripping and exposes a three-colored band, part of which is luminous so that indication is given under all conditions. Nontrip-free action is provided by a specially designed "shunting" switch. Compact and light in weight, the breaker will carry at least 120 per cent of rated current continuously, and trip ultimately at 138 per cent of rated current in an ambient temperature of 25 degrees Cent. Usual requirements of shock, vibration, and motion regardless of the position of the mounting are met. The breaker is available in ratings from 5 amperes to 50 amperes, with standard AN mounting dimensions, and is suitable for 30-volt direct-current systems.

Fan-Cooled Gear Reducer

INCORPORATING A fan-cooling system which permits a reduction in size of unit required for a given horsepower output, a new worm gear reduction unit, known as Speedaire, has been announced by Cleveland Worm & Gear Co., 3249 East Eightieth street, Cleveland 4. Of double-wall construction, the cooling system provides an air passage completely enveloping the oil reservoir in which the gear operates. The inner housing wall forming the oil reservoir is deeply finned on the air side, resulting in an increased heat-dissipating surface. Located on the coupling end of the worm shaft, an exhaust fan draws air at high velocity through the space between the housing walls from a grille at opposite end of unit. It is designed to operate in either direction of rotation. The operating temperature is lowered by this high velocity air stream, giving the unit a greatly increased load-carrying capacity.



Aircraft Autotransformers

ANNOUNCED BY General Electric Co., Schenectady, N. Y., two small size autotransformers have been designed to furnish alternating-current voltage for operation of 400-cycle aircraft instruments. They are light in weight, the 6 volt-ampere unit weighing 2 ounces and the 12 volt-ampere weighing 3 ounces. Dimensions are approximately 1¾ x 1¼ x 1 1/16 inches. Both operate over a frequency range from 380 to 420 cycles. These units can be applied on planes having a 110-120 volt, alternating-

CLEANS SANDY SEA WATER FOR INVASION-SUPPORT CRAFT

"Monel over Monel" wire cloth strainer gives double service

INTAKE WATER STRAINER, *4" OD x 12" long, used on Army and Navy craft operating near shore. 50-mesh Monel wire cloth cylinder, with brazed seam, is brazed on the backing of perforated Monel sheet, described below.*

The water sucked up near shore to cool engines of small craft carries silt and sand that pound at strainers.

In addition, the water itself subjects the strainers to the combined attack of corrosion and erosion.

Two years was considered a reasonable life for strainers until the "Monel over Monel" type shown here was developed. Such strainers have given over *five* years of service with little sign of wear . . . are so much more durable that they were adopted for use on Navy patrol boats, tank lighters, Army and Navy landing craft and rescue boats.

Made by the Gross Mechanical Laboratories, Baltimore, Md., these *double*-life *double*-strainers consist of fine Monel wire cloth backed up by perforated light gauge Monel sheet.

Monel wire cloth has the strength and toughness to resist abrasion and erosion, together with high resistance to corrosion by sea water and brackish waters. Monel is formed readily and is brazed and joined to other materials without difficulty, thus permitting speedy economical fabrication.

Today all prominent weavers produce Monel wire cloth for war requirements and for essential production and maintenance needs where Monel's rare combination of properties is needed.

THE INTERNATIONAL NICKEL COMPANY, INC.

67 WALL STREET **NEW YORK 5, N. Y.**

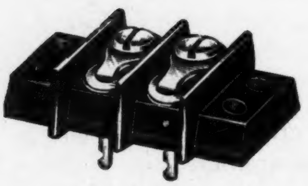
PERFORATED MONEL SHEET STRAINER, *used alone as strainer or with Monel wire cloth assembly where finer screening is required. Made from .031 Monel sheet, spot welded, with ends of tin-coated brass.*

NICKEL INCO ALLOYS

MONEL • "K" MONEL • "S" MONEL • "R" MONEL • "KR" MONEL • INCONEL • "Z" NICKEL • NICKEL • *Sheet...Strip...Rod...Tubing...Wire...Castings*

current source available, this eliminating the need for additional inverters to supply, from the direct-current source, the correct alternating-current voltage. With sufficient ventilation the autotransformers will operate at any altitude, and while rated at 120-volts input, are suitable for operation over a voltage range from 110 to 130 volts.

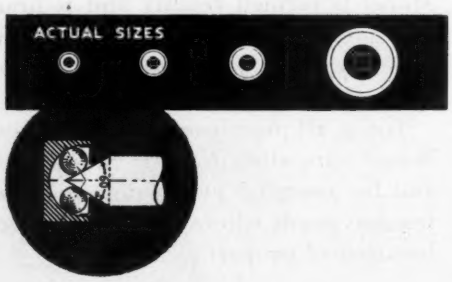
Barrier Type Terminal Strip



DUE TO THE demand for a barrier type terminal strip with facilities for connections both above and below the mounting surface, Howard B. Jones Co., 2460 West George street, Chicago 18, has designed its Y-type terminal which mounts on the company's standard barrier strips. This permits a screw connection above the panel and a solder connection below. Made for Nos. 140, 141 and 142 barrier strips, the Y-type terminal mounts securely in the block and is held by a screw so that the connecting wire comes in direct contact with the terminal. According to Navy specifications 17P4 the body is bakelite and the terminal is tin-plated brass. Being of sufficient height, the barriers give desired creepage distance and prevent shorts from frayed wires.

Pivot Type Ball Bearings

NOW AVAILABLE from Miniature Precision Bearings, Keene, N. H., are pivot type ball bearings in sizes ranging from 2 to 10 millimeters outside diameter. They are offered in beryllium, stainless or chrome steel as required

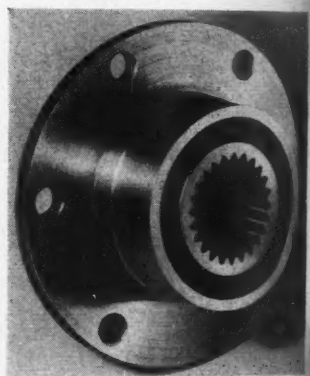


for specific applications. Bearing races are machined from solid bar stock and finished on raceway and exterior surfaces. Each bearing is equipped with four balls of the same material as the cup, and fitted with a retaining cap.

Relays for Circuit-Switching

TWO NEW RELAYS designed by Potter & Brumfield Mfg. Co., 103 North First street, Princeton, Ind., for circuit switching in radio equipment and other similar devices, have unlimited uses in many other fields, including aircraft installations. For applications where size and weight are important, Type KR relays are sturdy in construction. Where current is not too limited the direct-current types can be adjusted to withstand vibration encountered in most aircraft applications. They can be provided with mechanism for easy adjustment of armature spring where a critical pull-in value of current or voltage is required. Unless otherwise specified, contacts are fine silver, rated at 3 amperes, 110 volts, 60 cycles, noninductive. Coils available for voltages up to 110 volts, 60 cycles, or 60 volts direct current. Contact arrangements are up to and including double-pole, double-throw. This type is 1 11/16 inches long, 1¼ inches high and 1 3/16 inches wide, and weighs 1¾ to 2 ounces. The second series, Type KL, has approximately twice as much coil space and a larger number of poles, and will operate on values as low as .2 watts. Contacts are also fine silver and are rated at 3 amperes 100 volts, 60 cycle noninductive. Coils are supplied for voltages up to 220 volts, 60 cycle or 110 volts, direct current. Contact arrangements are up to and including four-pole, double-throw. The dimensions for this type are 1 13/16 x 1 13/16 x 1 13/16 inches.

Splined-Shaft Coupling

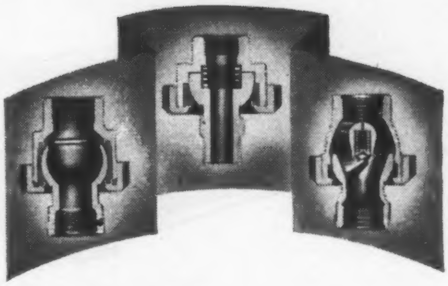


PRODUCED FOR various loads, a new splined-shaft coupling is being manufactured by Bushings Inc., 3442 West Eleven Mile road, Berkley, Mich., to stop the transmission of noise, absorb any shock and impact and to compensate for misalignment between drive and driven shafting. The new coupling uses synthetic rubber as the insulating medium to permit operation in the presence of lubricant, and consists of a splined inner sleeve, a flange and separating bushing of the synthetic. The same general design with either keyway or splines, as illustrated can be utilized for handling any load by the use of proper hardness and thickness of the rubber or synthetic. The company in all cases applies the rubber or synthetic to parts furnished by the customer to produce the finished couplings.

Floating Cage Type Speed Nut

FOR BLIND attachment the new self-locking Speed Nuts developed by Tinnerman Products Inc., 2085 Fulton road, Cleveland 13, are available in two styles: A6939 of brass and phosphor bronze for use with standard 6-32 machine screws, and A5939 of spring steel for use with standard 6Z sheet metal screws. Both styles fit panel thicknesses from .062-inch up and require only one clearance hole of .171-inch diameter for attaching. Approved by Army Air Forces, the new cage nut has a wide range of appli-

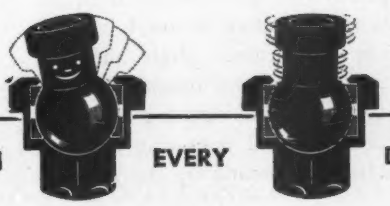
The combined ball-and-swivel action makes this a truly universal joint for fluid-conveying pipes...with movement in every direction. And the consistently satisfactory performance of Barco Flexible Joints in every field of industry and transportation for thirty years past has made them a favorite the world over. Barco Manufacturing Co., Not Inc., 1806 Winnemac Avenue, Chicago 40. Ill.



BARCO FLEXIBLE JOINTS

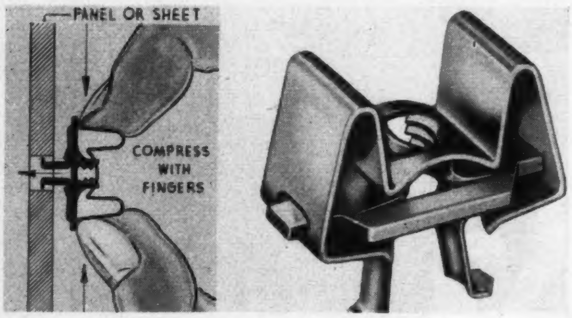
Not just a swivel joint...but a combination of a swivel and ball joint with rotary motion and responsive movement through every angle.

In Canada: The Holden Co., Ltd., Montreal, Canada



"MOVE IN EVERY DIRECTION"

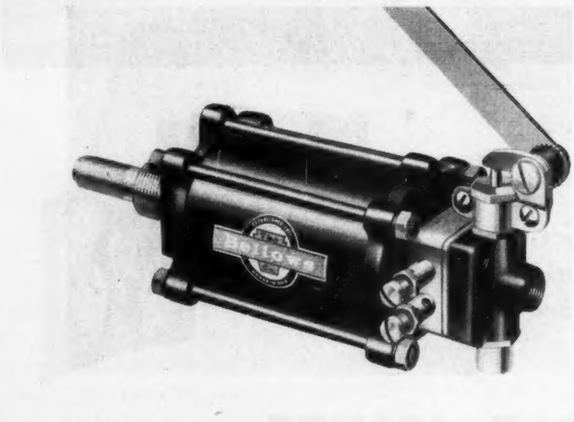
cations, and can be made in larger screw sizes to meet individual requirements. No special tools are needed to install the fastener. The cage is compressed, as shown in the accompanying illustration, and the legs are inserted into the clearance hole. These spring apart when pressure is released, the turned-up ends clearing the back



side of the panel. The nut will hold against the force of inserting the screw and the screw-tightening torque, and the floating nut within the cage allows for any misalignment of clearance holes.

New Type Air Motor Offered

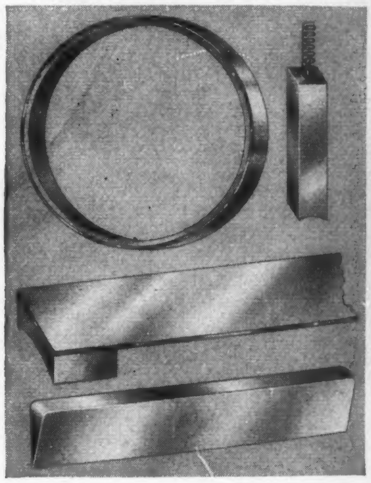
DEPARTING FROM conventional air cylinder design, Model BM air motor introduced by The Bellows Co., Akron, O., is a complete, self-contained power unit, with integral valve, operating lever and speed controls, permitting full control over all phases of operation. The motor operates on any air pressure up to 175 pounds per square inch. Power thrust is approximately five times operating air-line pressure, and air consumption is low. The air motor is available in strokes ranging from 1½ inches to 48 inches. Adjustable to any angle, the valve operating lever is easily synchronized by simple linkage,



cam or solenoid controls, to the machine cycle for automatic operation. Dual exhaust throttle controls permit speed regulation of the piston rod in either or both directions. The air exhaust streams may be used to eject parts from holding devices or to clean chips and dirt away from the work area. Many hand operations such as feeding work to tools, tools to work, opening and closing of vises and holding fixtures, and operating clutches and latches, can be made fully automatic by the application of this air motor. It also provides a source of fle[...] power for various staking, clamping and embossing [...]erations.

Silver in Electrical Equipment

FOR APPLICATION in equipment such as searchli[...] and airplane turrets, and in various precision instrume[...] and control equipment where maximum conductivity [...] minimum contact resistance are required, Gibson Elec[...] Co., 8355 Frankstown avenue, Pittsburgh 21, is prod[...]ing various current collector devices made of silver [...] silver alloys. Small and intricate commutator segme[...] of various shapes for numerous applications are ma[...] within close tolerances, with maximum hardness [...]



wearing qualities of both silver and silver alloys bei[...] maintained. Shown in the accompanying illustration a[...] a few typical parts being produced to meet the requir[...]ments of electrical manufacturers for special applicatio[...]

Pressure-Sealing Tapes

PRESSURE-SEALING tapes for identifying, codi[...] labeling, masking, splicing, insulating, protecting, etc., a[...] being offered in the Filmonize line of International Plasti[...] Corp., Morristown, N. J. This line consists of transpare[...] tapes, single and multicolored tapes, printed (numerica[...] alphabetical, or specially designed combinations), lumi[...]nous, electrical and masking tapes in widths ranging fro[...] ½ to 18 inches. Having high tensile strength, the tape[...] are weatherproof and are unaffected by extreme tempera[...]tures, sea air, salt spray, chemical fumes or war gase[...] Acetate backing reduces curl-back and telescoping, a[...] ends stretching and shrinking.

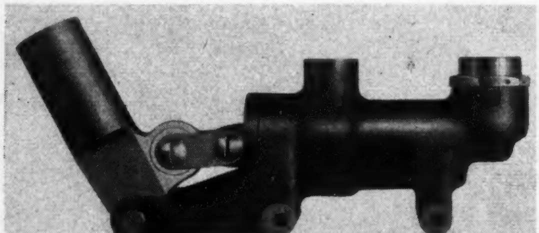
Ribbon-Rectangular Magnet Wire

AVAILABILITY OF ribbon-rectangular magnet wire i[...] shapes as thin as .004-inch has been announced by Ge[...]eral Electric Co. This is one-fourth the size previously [...] considered the low limit for thicknesses of this wire. [...]

is smooth, strong, flexible, and able to withstand high-speed winding without damage to insulation, offering many possibilities for application in electronic devices and other electric components. Because of the thinness of the wire it can be applied where round wire previously had to be used. In addition it will substantially increase the winding space factor and may be used in place of larger-size, rectangular magnet wire to increase magnetic effect or reduce coil size.

Hand-Operated Hydraulic Pump

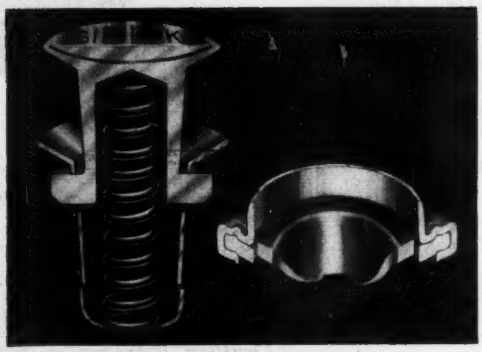
PARTICULARLY suitable for installations in which small quantities of fluid at high pressures are required, a new hand-operated hydraulic pump is being offered by Pesco Products Co., 11610 Euclid avenue, Cleveland 6. The rated capacity of this double-acting, differential, piston type pump is one-eighth gallon per minute at 20



cycles per minute, with 1500 pounds per square inch discharge pressure. Spring-loaded ball check valves provide unidirectional fluid flow. The needle bearings at the lever and pivot minimize operating torque. Packing is suitable for operation with mineral oil. Weighing only 2.4 pounds, the pump is lubricated by fluid passing through.

Cowling Fastener Offered

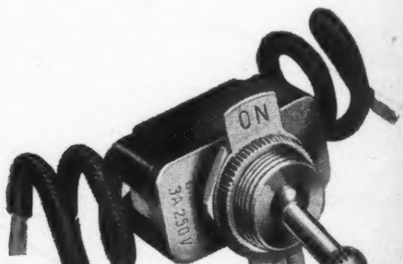
DESIGN AND construction improvements have been made in fasteners for cowlings and panels produced by the Simlok division of the Simmons Machine Tool Corp., Albany, N. Y. One of the improvements includes the



fastening and unfastening of the unit with a quick quarter-turn. When unfastened the stud is self-ejecting, permitting quick inspection. The fastener is available with three types of studs—flush head, oval head and wing h[ead] —and comes in three sizes. For assembling curved sh[eets] such as engine cowlings, the tapered design of the fa[s]tener makes it particularly valuable. The fastener eli[m]inates side play and holds end play or spring deflecti[on] to a maximum of .008-inch or just enough to lock t[he] fastener. It is constructed with a hollow inner one-pie[ce] housing integral with locking rings, and is casehardene[d] to eliminate wear. A long-travel helical spring ejects t[he] fastener when unlocked, and an outer housing retains t[he] spring and inner housing as a complete assembly. P[er]manent installation, if desired, is possible with a locki[ng] ring. To lock, the two sheets are brought together so t[he] stud enters the receptacle. Then the stud is depresse[d] against the spring action and turned until it rides und[er] the cam surfaces. It locks in a seat in the cam. Th[e] new cowling fastener is being used primarily in the avi[a]tion industry and has many potential postwar application[s].

Small, Thin Toggle Switch

FOR APPLICATION on small tools, appliances and offi[ce] machines, a new small, thin toggle switch has been de[]veloped by McGill Mfg. Co. Inc., Valparaiso, Ind. Co[n]struction of the switch is vibration and dustproof. Th[e] body is 1 inch long, ½-inch wide and ½-inch thick, an[d] will fit into a shallow opening. The accompanying illu[s]tration shows the switch in actual size. It is available i[n] three types: No. 25 single-pole, rated at 6 amperes 12[5] volts, and 3 amperes 250 volts. No. 27, three-way, rate[d] at 6 amperes, 125 volts. No. 28, two-circuit, no off posi[]tion, rated at 6 amperes, 125 volts. All three types ar[e]

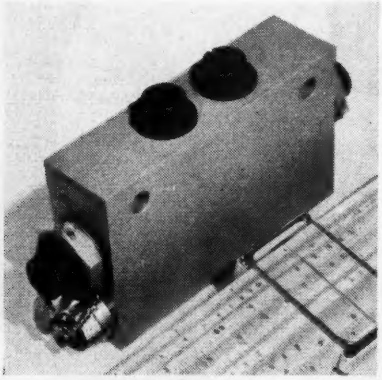


furnished with wire leads or solder lugs. In addition t[o] the above-mentioned uses, the switch has many othe[r] applications such as individual control of lights in Pul[l]man cars, passenger cars and buses, interior lights in auto[s], airplanes, camera projectors, etc.

Three Hydraulic Lock Valves

THREE NEW small size, lightweight, hydraulic lo[ck] valves are being produced by Adel Precision Produc[ts] Corp., Burbank, Calif., for use in aircraft to protect ma[]chine gun turrets, landing gear and other controls again[st] operation failures due to leakage. The new valves lo[ck] hydraulic fluid in cylinders or hydraulic motors until th[e] directional control valve is operated, thus maintainin[g] pressure in the actuating mechanism. Automatic in th[e]

operation they hydraulically lock the cylinder and are opened by pressure released by the selector valve. Operating parts are corrosion-resistant steel, and bodies are of dural bar stock, assuring freedom from porosity. A thermal relief valve is incorporated in each lock-valve cylinder port to bleed off excess pressure at from 1200 to 2000 pounds per square inch, depending upon requirements. The valves may be obtained for various tubing sizes and for operation ranges from −65 to 160 degrees Fahr. Type D12940, illustrated, weighs 1.18 pounds and measures 4 x 2 7/16 x 1¼ inches.

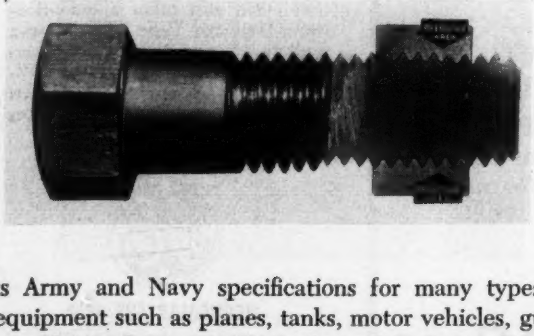


Three Peelable Stickers

ACCORDING TO a recent announcement of Avery Adhesives, 451 East Third street, Los Angeles, three of their peelable stickers now are available in stock sizes. These stickers are applied without moistening, stick to any smooth metal, glass, plastic or wood surface, and are easily peeled off when desired.

Self-Locking Nuts Available

NOW BEING MADE in fourteen bolt sizes, from ¼-inch to 1½ inch, with national coarse or national fine thread, the self-locking nut of Stover Lock Nut & Machinery Corp., Easton, Pa., is being produced under precision methods to insure uniformity in size and performance. Designed to withstand severe vibration, the nut meets Army and Navy specifications for many types of war equipment such as planes, tanks, motor vehicles, guns, and on ships of varying sizes, from P-T boats to submarines and battleships. The nut acts as a spring which clutches the bolt, this being achieved by subjecting the nut to the company's compressing process under which the upper part of the threaded portion is made slightly elliptical while the lower threaded portion remains circular. It can be screwed on a normal bolt about half way by hand, and then requires the use of a wrench. When the threads are on the bolt, the nut locks itself by utilizing the elasticity of the metal, steel or bronze. It is in one piece, and can be made to any torque required.



Injection-Molded Nameplates

TO MARK STEAM and diesel engines, injection-molded nameplates of Tenite are being produced. The nameplates are molded by Remler Co., Ltd., San Francisco, of Tenite produced by Tennessee Eastman Corp., Kingsport, Tenn. Unlike their predecessors, the nameplates will not corrode and outlines will remain distinct—no polishing being necessary. A sharp contrast with the metal is provided by one color being an inherent part of the plastic and another being furnished by paint. Because of the high visibility of the new nameplates, it is felt that Tenite will serve this purpose after the war as well. The plates are fastened to the engines by means of machine screws, and serial numbers and other data are stamped on with metal stamping dies.

High Heat-Resistant Plastic

SAID TO BE the first plastic ever developed that can hold its shape and strength in boiling water and yet be molded by the fastest methods, the new thermoplastic known as CEREX opens up a new field of industrial and household applications by virtue of its ability to withstand sterilization. Produced by Monsanto Chemical Co., St. Louis, the new plastic already has found wide use in war work, particularly Radar, radio and other military electronic equipment where light weight, suitable electrical and heat-resistant properties are required. It also is being used for surgical instruments, aircraft instruments and other war applications. Cerex is readily moldable in standard molding machines, and combines high resistance to heat with resistance to strong, corrosive chemicals, excellent insulating properties, high rigidity and strength.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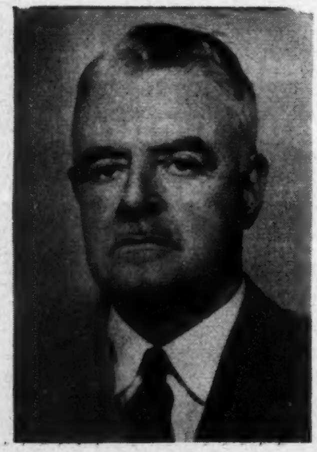
Men of Machines

RECENT ANNOUNCEMENT has been made by The Addressograph-Multigraph Corp., Cleveland, of the promotion of Lester F. Mitchell (left) to manager of engineering and of Franklin E. Curtis (right) to chief engineer of the company. Both men started by learning the toolmaker's trade. Previous to his appointment as manager of engineering Mr. Mitchell had been chief engineer, the post he had held since 1932 when he joined the corporation. Prior to that he had been connected with National Cash Register Co., Dayton.

Mr. Curtis, who succeeds Mr. Mitchell, became associated with Addressograph-Multigraph after spending several years in engineering work following World War I.



BEGINNING HIS professional career in the automotive field as an apprentice, Ellery R. Fitch has reached the top as director of engineering of Bendix-Westinghouse Automotive Air Brake Co., Elyria, O.—the position to which he recently was appointed. Immediately upon graduation from Syracuse university in 1912 as a mechanical engineer, he joined the Westinghouse Air Brake Co. as a special apprentice. Later he served successively in a variety of commercial, district and research posts with this organization, until 1938. At this time he became connected with Bendix-Westinghouse Automotive Air Brake Co., accepting the assignment as chief engineer and fulfilling the duties of this position until his appointment as director of engineering. Mr. Fitch is a member of the Air Brake association, Society of Automotive Engineers and the Army Ordnance association. During World War I he served as a Lieutenant with the United States Naval Reserve. He was born in 1889 in Elmira, N. Y.

FORMERLY CHIEF engineer with the Oldsmobile division of General Motors Corp., Harold T. Youngren has been appointed director of engineering development of Borg-Warner Corp. His previous engineering experience began in 1910 and included positions with Allis-Chalmers, Westinghouse, Harley-Davidson, and the Pierce-Arrow Motor Car Co. During World War I he was employed by Curtiss Aeroplane & Motor Corp., at Buffalo and Garden City, N. Y., as an experimental design engineer on airplanes and engines. He then joined the Studebaker Co. as executive engineer in charge of passenger car chassis design. In 1929 he went to General Motors and in 1933 became chief engineer of the Oldsmobile division, the position he held at the time of his present appointment. One of the chief duties of his new post will be supervision of Borg-Warner research laboratories, and he will cooperate with the engineers of the corporation's divisions and subsidiaries. While the organization has labora-

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tories at Detroit and Rockford, Ill., Mr. Youngren will use Chicago as headquarters. As far as technical society affiliation is concerned, he has been an active member of the Society of Automotive Engineers for more than thirty years.

♦

George H. Clark, vice president in charge of engineering, The Formica Insulation Co., recently was elected a director of the Society of the Plastics Industry.

♦

Clifford T. Appleton has been appointed manager of the machine division of Rolled Thread Die Co., Worcester, Mass. For seven years previously he had been vice president of Leon J. Barrett Co., and before then was associated with Heald Machine Co. and Reed Prentice Co.

♦

Raymond W. Scheubel who had been a tool designer for Allied Process Engineers is now service engineer for Elastic Stop Nut Corp., Newark, N. J.

♦

Clarence A. Hubert, formerly in the engineering department of the Farm Tractor Division, International Harvester Co., Chicago, has been appointed chief engineer of the division.

♦

Paul C. Vyff has become connected with Battelle Memorial institute, Columbus, O., as research engineer Formerly he had been chassis designer, truck division, Dodge Division of Chrysler Corp., Detroit.

♦

A. C. Boock of Allis-Chalmers Mfg. Co., Springfield Works, Springfield, Ill., has become assistant chief engineer of the company. He previously had been chief design engineer.

♦

T. C. Kuhlman, previously connected with Miller Engineering Corp. as research engineer now is associated with Allis-Chalmers Mfg. Co., West Allis, Wis., as a consultant on precision castings.

♦

Ralph H. Upson has been made chief aeronautical engineer for a War Production Board research project at the College of Engineering, New York university. In this research project it is sought to utilize wind as a source of electric power on a large scale.

♦

F. A. Dobson has become chief aeronautical engineer at Aeronautical Products Inc., Detroit. He had been project engineer in charge of glider design for Waco Aircraft Co., Troy, O.

♦

O. G. Blocher has been transferred to the Stout Research Division of Consolidated Vultee Aircraft Corp., as chief engineer. He had been connected with the Stinson division, Wayne, Mich., as executive engineer.

♦

R. C. Henshaw is the new chief engineer responsible for all design and development for Lord Mfg. Co., and Leon Wallerstein Jr. will have charge of the development laboratory.

Patent Courts

(*Concluded from Page* 96)

though it would seem over the long period of years that the burden is on the defendant to show that there was no invention, I have to reach the conclusion, as a result of the study of Supreme Court decisions during the last few years and as I interpret the decisions of the United States Circuit Court of Appeals for this circuit, that that burden, in effect, shifts, and it becomes imperative for the plaintiff to show that his contribution to the art was substantial.

Patentee's Attorney: On that point, your Honor, it leaves the inventor in a very precarious situation.

The Court: It certainly does.

Patentee's Attorney: The Government issues a patent, takes the inventor's fees. The inventor spends money for a patent attorney. And if his invention has utility and novelty he has the burden of then proving that it amounts to invention. It puts him in a position where large corporations can adopt his invention even though it has utility and novelty and throw a further burden upon him of coming into court and spending a large sum of money to establish that he actually has made what amounts to invention. Now, of course, that makes it a great hardship upon any individual.

The Court: That is the situation that every plaintiff in a patent suit has to face when he comes into court. It is not for a trial court to do other than to conform to the expressed authority of the highest courts.

The judge in the dialogue quoted was, prior to his appointment to the district court, an attorney engaged in patent practice, and would be representative of those qualified to be selected for the proposed court of patent appeals. A check on his patent cases over a three-year period reveals that, of 11 patents, he held one valid and infringed and 10 invalid or not infringed, making a "lost" percentage of more than 90 per cent. This illustrates the fact that the adoption of a single court of patent appeals will not alone settle our patent problems if the judges who are to be selected declare 90 per cent of the patents invalid or not infringed. This high percentage of patent mortality will drive inventors to keep their inventions secret. Of vitally more importance than the adoption of a single court of patent appeals is congressional legislation defining a standardized definition of what constitutes *patentable invention*, as discussed in Machine Design, February, 1944. Such a standardized definition would eliminate the predicament in which judges find themselves today where they are forced to disregard the presumption that the patent is valid for fear of being reversed should they hold the patent valid and infringed. If all courts were geared together by having the same test for patentable invention, then the need for materially changing our system by establishing the single court of patent appeals would appear secondary.

Stepped extrusions, in some cases, provide economy in material cost and in manhours required to machine finished parts such as the spar caps used in some aircraft designs.

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CAPACITY

400 gal. per hr. at 2500 rpm.

6 to 35 lbs. pressure per sq. in.

Weight only 2¾ lbs.

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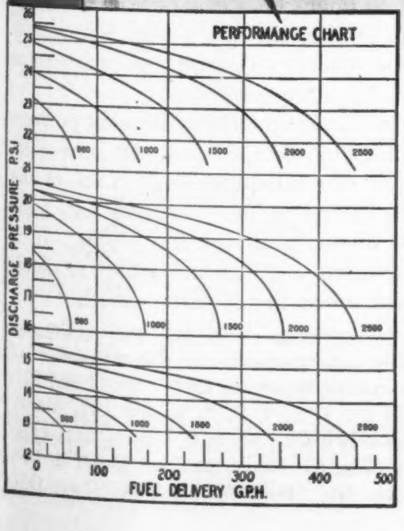

If you can use higher fluid pressure you can save time. This pump has a capacity of 400 gal. per hr. at 2500 rpm. Suppose you save only 2 man-hours a day per pump; that means a saving of from \$40 to \$60 a month.

Higher pressure range may also help you. This G-9 runs from 6 to 35 lbs. per square inch. Furthermore it weighs less—only 2¾ lbs. Its balanced type relief valve with shaft seals are dependable through extreme ranges of temperature. A precision built pump noted for its unfailing dependability, but only one of many in the ROMEC line.

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NON-PULSATING PUMPS... ACCESSORIES

Clutches

(*Concluded from Page* 106)

applied to the coil through two collector rings usually mounted on the field hub. Magnetic attraction, consequent to energization, draws the two members together with a positive and uniform pressure for development of the required driving torque. Upon interruption of the current, a spring disk or set of spiral springs separates the two members almost instantly, providing positive idling clearance. Magnetic clutch coils consume a nominal amount of current and may be wound for operation on any specified direct-current voltage.

In the clutch pictured in *Fig.* 11 the driving force is friction. There is, however, another type available employing serrated or toothed engaging surfaces and it is used where instantaneous engagement free of slippage is required. The serrated type, an application of which is shown in *Fig.* 12, can be engaged only at a standstill, but may be released while turning.

Because most types of machines require considerably

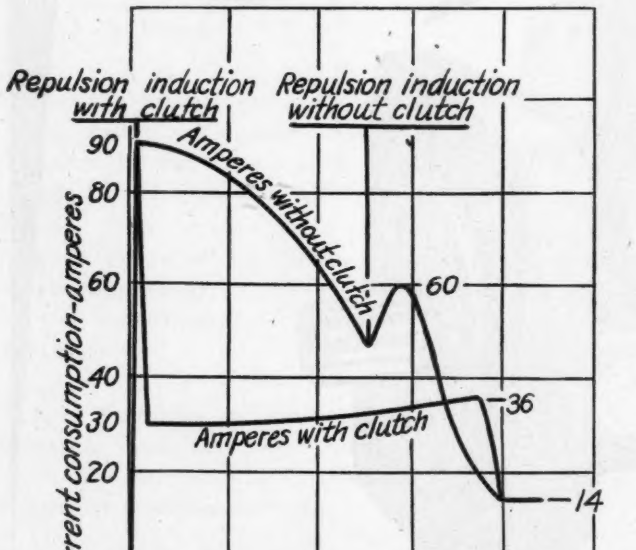


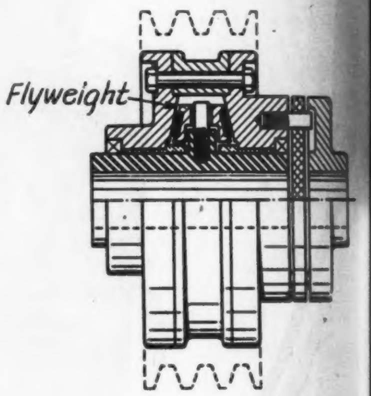
Fig. 13—Power consumption curves of comparison tests show the smaller amount of current needed to start motor when centrifugal clutch is used to couple motor and load

more power for starting than for operating after being brought up to speed, and many commonly used power sources such as electric motors and internal combustion engines produce relatively low torques at low speeds, over-powering of machines solely for the purpose of starting often results.

This brings into the picture a type of clutch which is meriting ever-increasing recognition in the machine design field—the centrifugal clutch. Interposed between driver and load, a centrifugal clutch engages only after a predetermined speed has been reached. Thus the driver is not called upon to pick up the load until it has developed a large proportion of its total torque.

Some idea of how much power can be saved during starting through the use of a centrifugal clutch is given by the curves in *Fig.* 13. While these curves apply specifically to an application using a mercury type of centrifugal clutch, they are nevertheless indicative of centrifugal clutch performance in general. The mercury type is simple in design, consisting mainly of a driving member or housing, an inner drum which acts as the driven member, segmented friction members disposed radially between the housing and inner drum, a cover plate and a

Fig. 14 — In this type of centrifugal clutch, engagement is effected when a number of flyweights are forced outward against the sloping walls of the outer clutch member



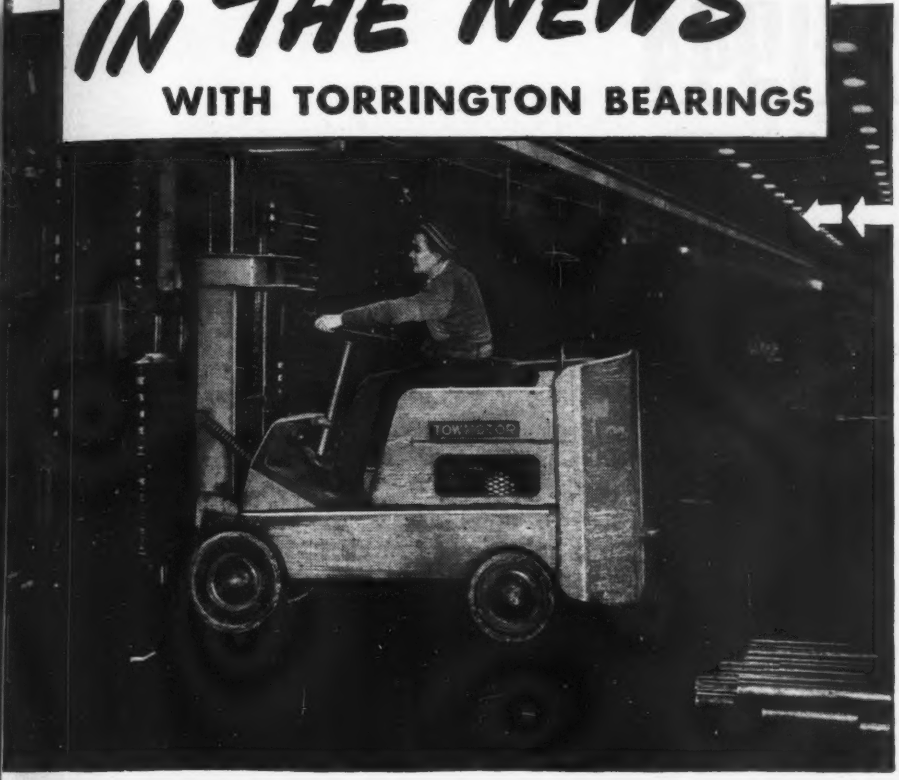
quantity of mercury. As the clutch speeds up, the mercury is thrown outward by centrifugal force and distributed evenly around the inner wall of the housing where it exerts a pressure on the clutch segments which in turn press against the inner drum and create the driving friction. Since the torque capacity of the centrifugal clutch varies as the square of its rotational speed, it is most effective at high speeds. Below 1000 revolutions per minute it must be built to a relatively large size.

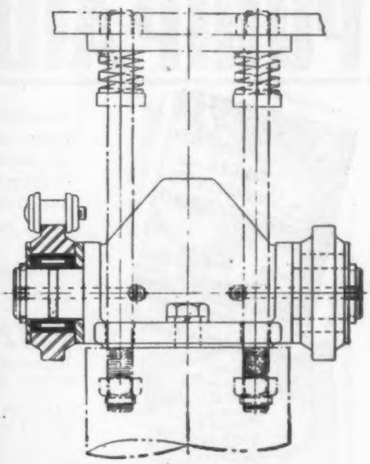
Another simple type of centrifugal clutch is that shown in *Fig.* 14. As will no doubt be apparent from the drawing, engagement is established when the flyweight is pressed by a sufficient amount of centrifugal force against the sloping walls of the outer member.

It will be seen from the foregoing that constant progress is being made in the clutch field, always with an eye toward smoother, better controlled coupling. Each of the clutch types has its special merits—each fits into some specific applications better than do the others. It is hoped that this discussion will aid the reader in determining the types best suited to his particular job.

Machine Design acknowledges with appreciation the collaboration of the following companies in the preparation of this article: Dings Magnetic Separator Co. (*Figs.* 11 and 12); The Hilliard Corp. (*Figs.* 1, 2, 4, 7, 8, 9, and 10); Kinney Mfg. Co. (*Fig.* 3); Link-Belt Co.; Mercury Clutch Corp. (*Fig.* 13); Morse Chain Co. (*Fig.* 5); Newaygo Engr. Co. (*Fig.* 14); Rockford Drilling Mach. Div., Borg-Warner Corp.; Twin Disc Clutch Co. (*Fig.* 6).

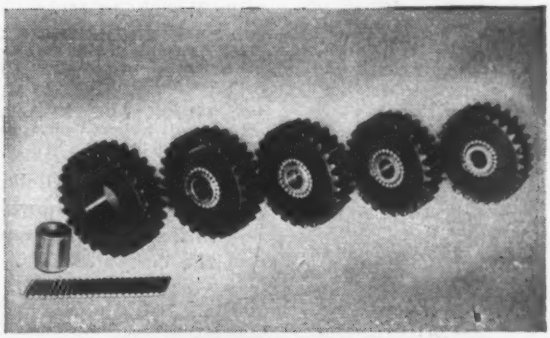
IN THE NEWS
WITH TORRINGTON BEARINGS



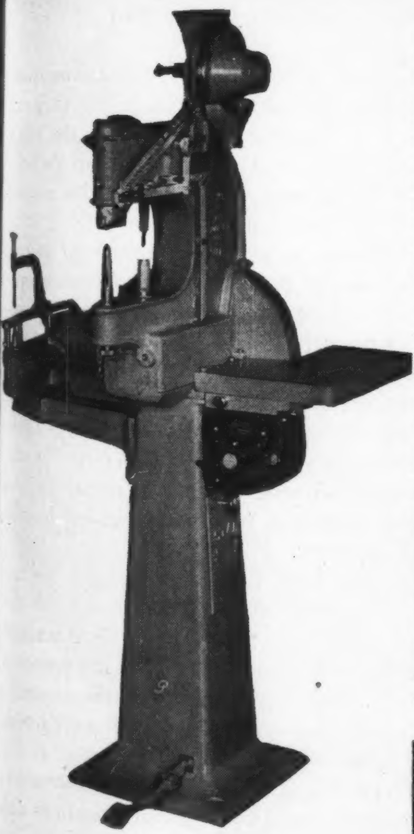


LIFT TRUCKS, manufactured by the Towmotor Corporation, are designed for the safe, rapid handling of multi-ton loads in around-the-clock service. To provide smooth operation in raising and lowering the load, Torrington NCS Needle Bearings are used on the hydraulic lifting cylinder crosshead. X-section clearly shows how compactness and high capacity of these bearings make them especially suitable for this interesting application.

MORE THAN 150,000 Jahco airplane starters have been manufactured by the "associates" of Jack & Heintz, Inc. Noted for their dependability and instant response to the starter switch, Jahco starters are now used exclusively in all large combat and bombing aircraft. Due to the necessity for compact design and light weight, Torrington LN Needle Bearings were selected for use in the gears shown at the right.



"OUT OF THE ORDINARY" bearing applications are an important phase of Torrington's Engineering Service. The specifications may call for a bearing of unusual size, or special design. Frequently a bearing previously engineered for one application can be adapted to meet the requirements of another. Man in foreground in photo at right is handling the inner race of a large double-row tapered roller bearing, a type widely used in heavy-duty mill applications.



DELINING & RELINING of brakes is done rapidly and efficiently in a series of automatic operations performed by this brake relining machine manufactured by the Star Machine & Tool Company. Front and rear plungers, which perform the delining and riveting operations, operate at nearly 300 R.P.M. To reduce friction to a minimum, special Torrington Roller Bearings are used for the plunger rollers.

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Vibration and Noise

(*Concluded from Page* 90)

measure acceleration, etc.

Choice between displacement, velocity or acceleration depends upon the application. Inasmuch as vibrating stresses in a machine are a function of acceleration, this characteristic is probably the most widely used. In heavy machines, with low-frequency vibrations, displacement measurements often are made.

Observation of vibration may be made in a number of ways. The vibration meter, *Fig.* 10, and the unit to the left of *Fig.* 11, indicate the root-mean-square of the total vibration. The complete assembly shown in *Fig.* 11, consisting of the meter at left and the analyzer at right, provides indications of the magnitudes of the separate harmonics comprising the vibration. Other types of equipment furnish records of the vibration spectrum in the same manner as the sound analyzer already described. Use of a direct-inking oscillograph, *Fig.* 12, provides a permanent record of the actual wave shape which can then be inspected and analyzed at leisure. Other forms of oscillograph furnish a visual indication of the desired wave shape on a screen. The portable unit illustrated in *Fig.* 1 traces a record on a celluloid ribbon.

Localizing Sources of Vibration

Compared with noise observations, vibration measurement and analysis have the great advantage that they can be highly localized by placing the pickup at the exact point desired, as in *Figs.* 1 and 10. The pickup also can be used to explore for the source of vibrations by moving over the surface of the machine.

Discussion of vibration measurement would be incomplete without reference to the usefulness of stroboscopic devices in the study of large-amplitude vibration. Illumination of vibrating parts by timed flashes of light of exceedingly short duration can create the illusion of slow motion or stopping, depending upon the relation between flash frequency and vibration frequency. Thus by "slowing down" the vibration until it can be followed by the eye the simultaneous behavior of several points can be observed, supplementing the information given by the vibration meter or analyzer.

It is hoped that this brief discussion will aid the designer in deciding what type of equipment best meets the needs of his problems. Frank discussion with instrument manufacturers of the exact requirements of any proposed new equipment will result in the final selection of equipment which will yield the maximum overall satisfaction.

Part II of this series will include a discussion of the suppression of vibration and noise, with particular reference to isolation through the use of special mountings and insulating materials.

MACHINE DESIGN acknowledges with appreciation the cooperation of the following in the preparation of this article: American Standards Assoc. (*Figs.* 5 and 6); The Brush Development Co. (*Figs.* 9 and 12); General Electric Co. (*Figs.* 7 and 10); General Radio Co. (*Figs.* 3, 4 and 11); Western Electric Co. (*Fig.* 8); and Westinghouse Electric & Mfg. Co. (*Fig.* 1).

WALDES TRUARC presents a significant advance in retaining rings. It spreads or contracts without distortion; always retaining its perfectly fitting circular contour.

For thrust-load fixing, and shaft and housing applications, Waldes Truarc provides distinct advantages over nuts and bolts or wedges and washers... it reduces dimension and weight ...saves material...cuts manufacturing time... simplifies assembly and dis-assembly.

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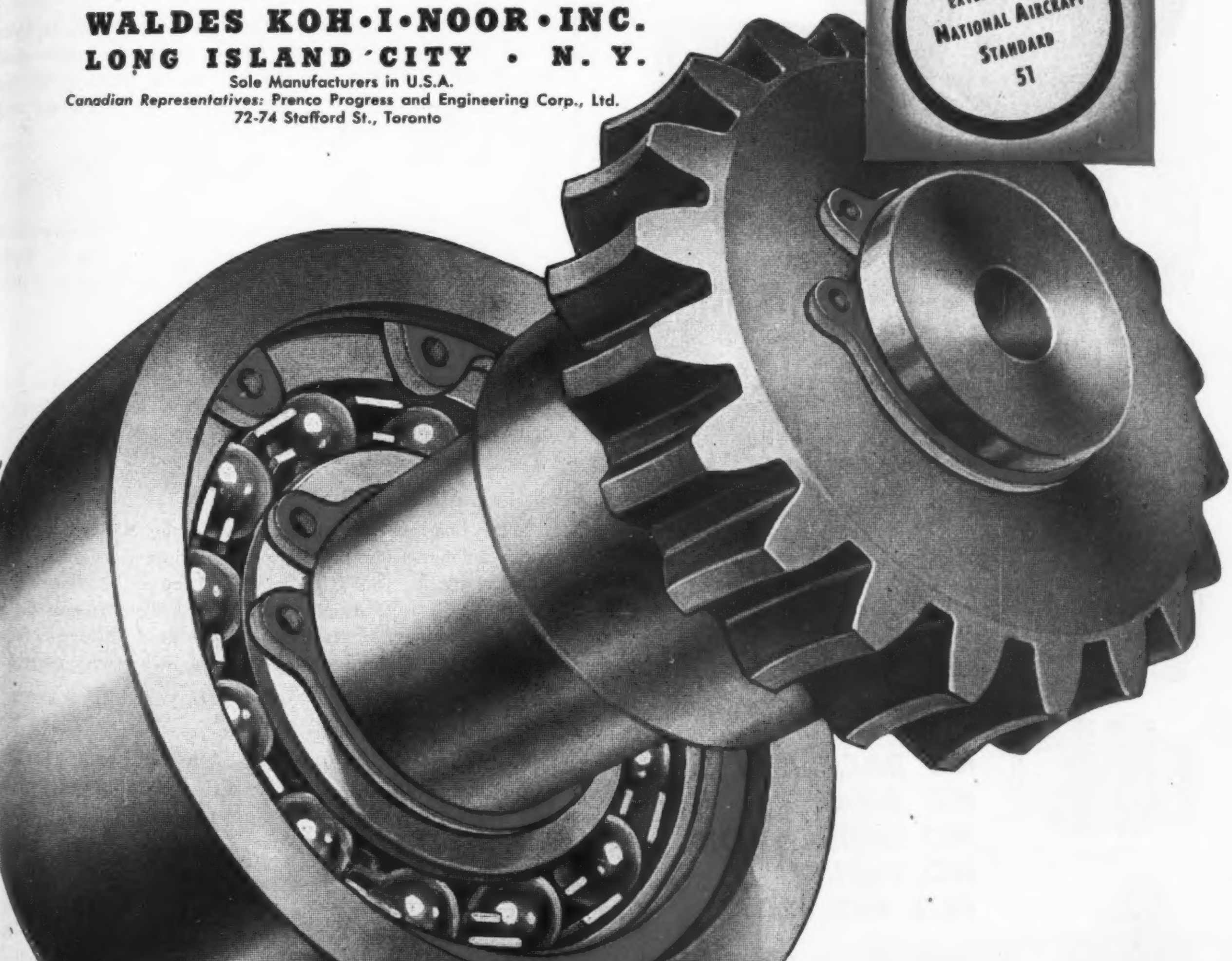
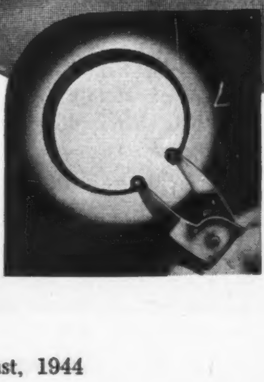




U. S. Patent
Re 18144

WALDES
TRUARC
RETAINING RING

BUSINESS AND SALES BRIEFS

PROMOTION of E. W. Stoner from Detroit district manager to sales manager in charge of industrial sales has been announced by Haskelite Mfg. Corp. He has been associated with the company for twenty-five years.

♦

Goodyear Tire & Rubber Co. has appointed R. D. Vickers in charge of plastics and chemical sales in the eastern region. Mr. Vickers, who will make his headquarters at 600 West Fifty-eighth street, New York, is a member of the newly created plastics and chemical sales division.

♦

Frank J. Brogan has been made special sales representative in Texas, Oklahoma, Arkansas and Louisiana for the Bunting Brass & Bronze Co. and will make his headquarters at Dallas.

♦

Appointment of the Geo. M. Prescott Co. to cover distribution of products in the Los Angeles district has been announced by Robbins & Myers Inc., Springfield, O.

♦

With headquarters in San Francisco, Thomas J. Kehane has been made commercial vice president in charge of Pacific Coast activities for Worthington Pump & Machinery Corp., Harrison, N. J. Formerly manager of New Jersey sales and assistant manager of the New York district office, Mr. Kehane will also be in charge of activities in territory covered by the Salt Lake City district office.

♦

Naming of Julian F. Gordon as assistant manager of sales has been announced by Stainless Steel division, Carnegie-Illinois Steel Corp., Pittsburgh.

♦

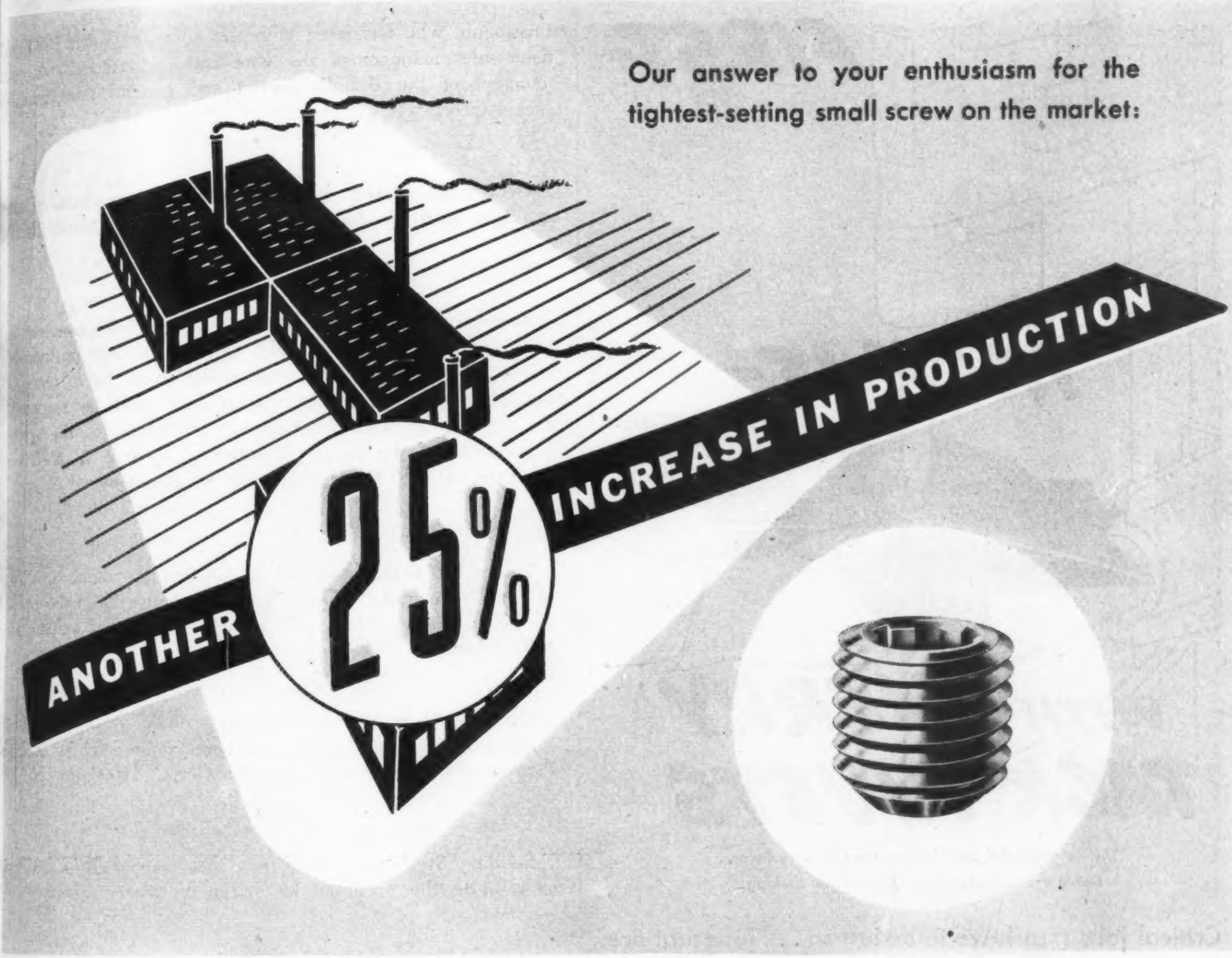
According to a recent announcement Charles A. Benz has become general sales manager of both Chicago Malleable Castings Co. and Allied Steel Castings Co., Chicago.

♦

Connected with the company for two years, Henry D. Engelsman has been named sales manager and technical service supervisor of Metallizing Co. of America, Chicago. Previously he had been sales and service engineer for DeVilbiss Co., Toledo, O.

♦

Advancement of A. G. Bussmann from assistant to the executive vice president to assistant to the president has been announced by Wickwire Spencer Steel Co. Mr.



We've had to keep on adding to our capacity for turning out Bristo Multiple-Spline Socket Set Screws. Recently, facilities were increased another 25%, representing hundreds of thousands more screws per week *in the wire sizes alone.*

The reasons why more and more manufacturers of electronic and electrical apparatus, airplane parts, photographic, scientific instruments and a multitude of similar products have adopted Bristo Screws include·

Assemby is *FASTER*, because of the easier transmission of rotary wrenching power.

Assembly is *EASIER*, because there is no slippage — the key is geared to the screw.

Assembly is *SIMPLER*, because Bristo screws easily reach hard-to-reach places.

Assembly is *TIGHTER*, because the screw can be turned farther without bursting or rounding out.

Disassembly is *QUICKER*, because the screw releases with a flick of the key — without socket damage.

Why "BRISTO" *Means* "TIGHTER"

No expanding pressure; the key pulls the screw around.

MULTIPLE SPLINE SOCKET · BRISTO SET & CAP SCREWS

The BRISTOL Company
MILL SUPPLY DIVISION

122 Bristol Road, Waterbury 91, Conn.

BRISTO MULTIPLE-SPLINE SOCKET SET SCREWS

GEARED TO THE KEY — FOR FASTER, EASIER, TIGHTER SETTING

Bussmann, who has been with the company since 1930, has been sales manager of the wire and springs divisions, sales manager of the Buffalo district and general sales manager successively.

♦

George W. Garvin has been given the position of sales manager, Aero Products Division, Talon Inc., Meadville, Pa.

♦

Promotion of Paul C. Roche to chief field engineer has been announced by Lord Mfg. Co. In his new position Mr. Roche will supervise field engineering work, be in charge of the home office sales engineering staff, and be responsible for delegating field engineering problems to the proper department at the plant. Transfer of George B. Noll from the Erie office to Chicago also has been announced.

♦

Under the management of E. C. Bolton, who is in charge of the district sales office, a new warehouse has been opened at 1200 Walnut street, Cincinnati, by Cutler-Hammer Inc., Milwaukee.

♦

Opening of a west coast branch office at 5655 Wilshire boulevard, Los Angeles 36, has been announced by Eclipse-Pioneer division, Bendix Aviation Corp., Teterboro, N. J.

♦

Scullin Steel Co., St. Louis, has named R. C. Geekie assistant to the president in charge of sales.

♦

Appointment of Edward T. Nahill as sales manager has been announced by Ace Mfg. Corp., Philadelphia, specialists in precision grinding, metal stamping and machine work. Mr. Nahill had been associated with General Electric Co.

♦

Construction on additions to the Neoprene plant at Louisville, Ky., has been begun by E. I. Du Pont De Nemours & Co. Inc., Wilmington 98, Del. Capacity will be increased fifty per cent.

♦

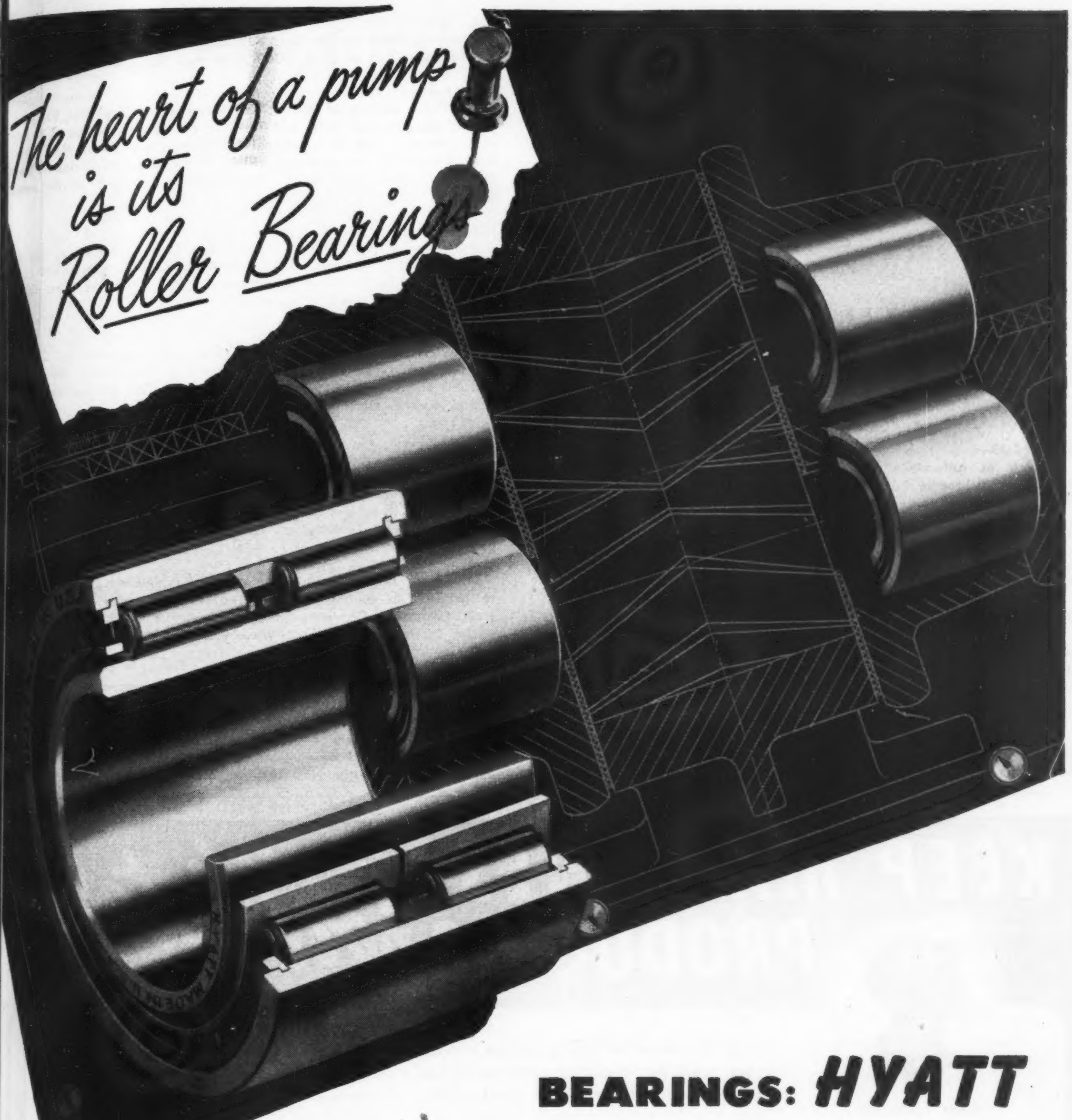
Purchase of New England Drawn Steel Co., Mansfield, Mass., has been announced by Bliss & Laughlin Inc., Harvey, Ill. The company will be operated as New England Drawn Steel division of Bliss & Laughlin Inc. and will be under the direction of Harold L. Sherwin, who continues as general manager with headquarters at Mansfield.

♦

A. G. Budd has been named mill supply products salesman for the Cleveland and Pittsburgh areas of The Bristol Co., Waterbury 91, Conn., and will make his headquarters at the Cleveland branch office located in the Engineers building.

♦

Five new engineering representatives have been appointed by Cannon Electric Development Co. They are: Wright Engineering, 6109 North Meridian street, Indianapolis 5; Franklin Sales Co., Central Savings Bank, Denver 2; Bruner



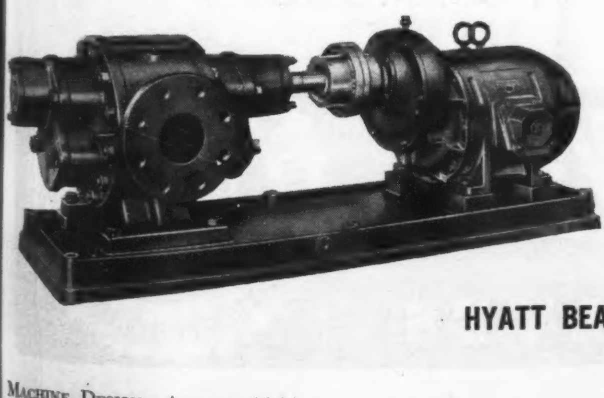
BEARINGS: *HYATT*

Rotary pumps that regularly handle extremely viscous liquids such as tar, molasses, asphalt and rosin, as well as liquids having lubricating value, for top efficiency must have roller bearings of large capacity and long life.

The selection of Hyatt Roller Bearings indicates that the manufacturer has done the utmost to assure the maximum of pump performance.

Whatever your bearing problem may be, call on Hyatt engineers. They bring much valuable help to you.

Worthington Rotary Pumps with double helical gears are Hyatt Roller Bearing equipped.



HYATT BEARINGS Division of GENERAL MOTORS CORPORATION
Harrison, New Jersey

Corp., 418 West North avenue, Milwaukee 12; Souther Sellers, 918 Union street, New Orleans 13; and Mountain States Engineering Co., 215 West Second, Salt Lake City 1, Utah.

♦

Moving of the firm to new and enlarged quarters at 1 East Fifty-third street, New York, and appointment of four new associates has been announced by Martial & Scull, industrial designers.

MEETINGS AND EXPOSITIONS

Sept. 13-15—
Society of Automotive Engineers Inc. National tractor meeting to be held at Schroeder hotel, Milwaukee. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Oct. 2-4—
American Society of Mechanical Engineers. Fall meeting to be held at Netherland-Plaza hotel, Cincinnati. Clarence E. Davies, 29 West Thirty-ninth street, New York, is secretary.

Oct. 5-7—
Society of Automotive Engineers Inc. National aeronautic meeting and engineering display to be held at The Biltmore hotel, Los Angeles. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Oct. 12-14—
American Society of Tool Engineers. Semiannual meeting to be held at Hotel Syracuse, Syracuse, N. Y. Adrian L. Potter, 2567 West Grand boulevard, Detroit, is executive secretary.

Oct. 13-14—
Electrochemical Society Inc. Fall meeting to be held at Hotel Statler, Buffalo. Colin G. Fink, 3000 Broadway, New York, is secretary.

Oct. 16-18—
American Institute of Mining and Metallurgical Engineers. Fall meeting of the Iron and Steel and Institute of Metals divisions to be held in conjunction with the National Metal congress, Public Hall, Cleveland. Convention headquarters will be at Hotel Statler. Frank T. Sisco, 29 West Thirty-ninth street, New York, is secretary of the metals divisions.

Oct. 16-20—
American Welding society. Meeting held in conjunction with the National Metal congress, Public Hall, Cleveland. Convention headquarters will be at Hotel Cleveland. M. M. Kelly, 33 West Thirty-ninth street, New York, is secretary.

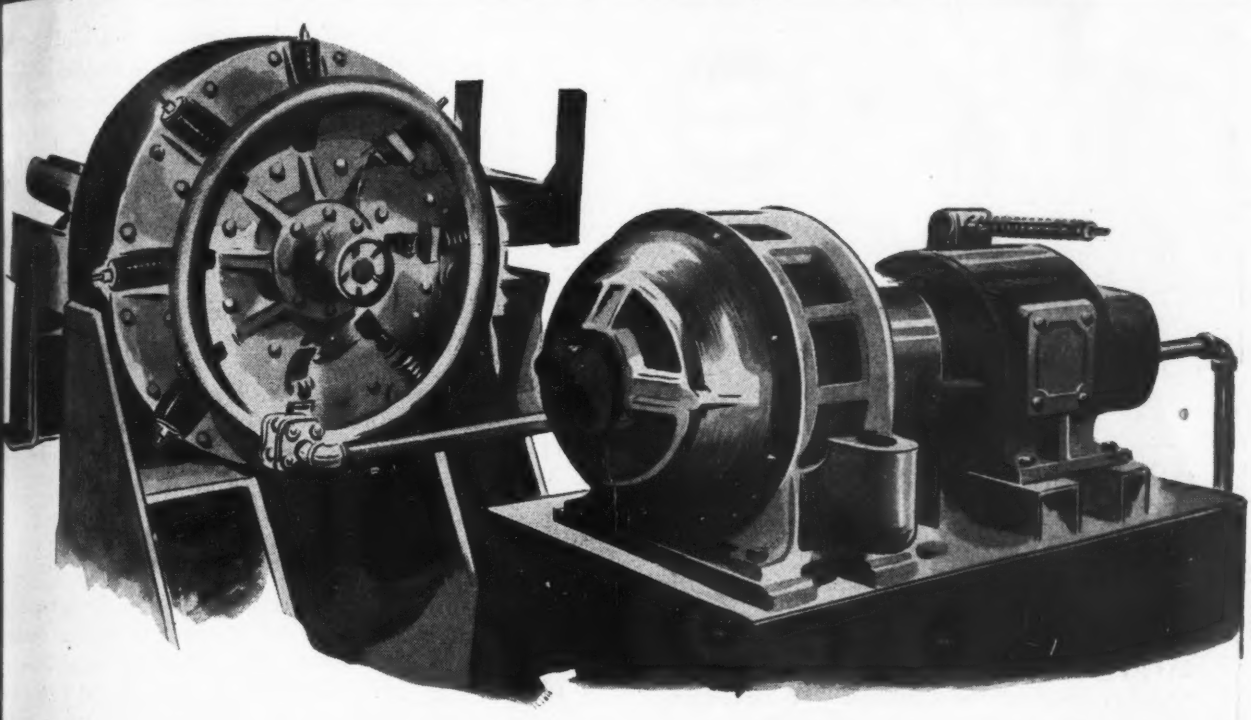
Oct. 16-20—
American Society for Metals. National Metal congress and war conference displays to be held at Public Hall, Cleveland. Headquarters will be at the Statler and Hollenden hotels. W. H. Eisenman, 7301 Euclid avenue, Cleveland, is secretary.

Oct. 17-20—
Society for Experimental Stress Analysis. Annual meeting to be held in conjunction with the National Metal congress, Public Hall, Cleveland. Convention headquarters will be at Carter hotel. Additional information may be obtained from headquarters of the Society at Massachusetts Institute of Technology, Cambridge, Mass.

Oct. 19-20—
American Industrial Radium and X-Ray society. Annual meeting to be held in conjunction with the National Metal congress, Public Hall, Cleveland. Convention headquarters will be at Hotel Hollenden. Philip D. Johnson, 25 East Washington street, Chicago, is secretary.

Oct. 19-21—
Electronic Parts and Equipment Industry conference sponsored by Association of Electronic Parts and Equipment Manufacturers, Sales Managers Club and the National Electronics Distributors association, to be held at Stevens hotel, Chicago. Additional information may be obtained from Charles Golenpaul, chairman of the publicity committee, care of Aerovox Corp., New Bedford, Mass.

What a Handshake these fingers pack



A story of a squeeze delivered by Hele-Shaw Fluid Power

BEFORE a tube is drawn through a die, the end is "pointed." Formerly it was simply bashed in by a process of hammering, a noisy operation that necessitated constant turning of the tube. Someone thought the job could be done better hydraulically. And did it.

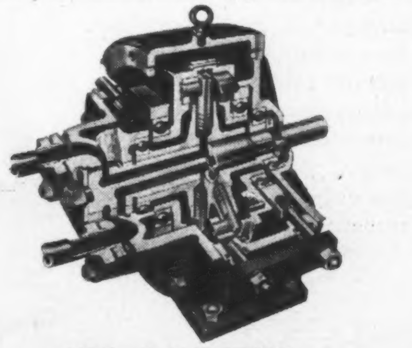
Now, in a "squeeze pointer" a cluster of "fingers" crushes the end of the tube to a near point in one mighty squeeze. A throw of a lever . . . and wham! Just one quick stroke, that's all. The pressure of the Hele-Shaw Fluid powered fingers is calculated so the end of the tube won't completely close—a neat example of pressure control.

Hele-Shaw Fluid Power does the job in seconds, saves time. Saves material because tube end wastage can be rigidly controlled. It's easier and quieter. Squeezing, lifting, tilting, pushing, pulling, and a variety of similar mechanical actions can often be improved by using Hele-Shaw Fluid Power. We're ready to work with you on future applications.

THE

Hele-Shaw

Fluid Power Pump



OTHER A-E-CO PRODUCTS:
TAYLOR STOKERS, MARINE DECK AUXILIARIES, LO-HED HOISTS

AMERICAN ENGINEERING COMPANY

2502 ARAMINGO AVENUE • PHILADELPHIA 25, PA.

Trimetric Drawing

(*Concluded from Page* 140)

As the copy board moves forward and backward funnel raises and lowers, permitting the camera to record only that portion of the negative which is intersecting plane parallel to the plane of the camera at any given moment. In this way it scans the entire drawing and records the axonometric projection on the sensitized paper which then is developed in the same manner as an ordinary photograph.

Prints of the various parts then are returned to the drafting room where they are positioned properly in the trimetric drawing and traced. All that remains for the draftsman to do is fill in the other two planes and connecting members for which no prints have been made. Even this work has been simplified considerably by use of the four new instruments which were developed along with the machine.

New Instruments Described

As shown in *Fig.* 3, the trimetric scale is triangular shape with an A face, a B face and a C face for measurements on the A, B and C axes and of course all lines parallel to these axes. Each face is graduated in the four scales 1:1, 1:2, 1:4 and 1:8, thus providing a single scale for the majority of fractional measurements. A similar instrument is graduated 1:1, 1:10, 1:30 and 1:60 for decimal measurements.

Shown in *Fig.* 4 is the trimetric template which provides three series of ellipses, one series for each of the three principal faces of the trimetric cube. The fifteen ellipses in each series are in sizes ⅛-inch to 1 inch, increasing in increments of 1/16-inch.

Designed for drawing ellipses larger than one inch, the trimetric underlay, *Fig.* 3, consists of a trimetric cube on which ellipses have been concentrically arranged eight to an inch on each of the three planes or faces, with the three outer ellipses graduated in degrees. In use it is slipped under the drawing to the proper location and the desired size of ellipse is traced off. It can be used also to scale nontrimetric lines (lines in the three principal trimetric planes not parallel to the coordinate axes), because a diameter drawn on a principal plane of the cube at any angle is intersected "in scale" by the concentric ellipses.

Similar to the ellipse template, with outer shape and notation of the coordinate axes identical, is the trimetric protractor. For any of the twelve trimetric positions, it can be used for angular measurements on the three principal planes of the cube. The degree graduations on the protractor are extended beyond the customary semicircle to one of the coordinate axes so as to provide reference axes A, B or C and the major axis in each of the three principal planes.

As a method of production illustration, trimetric drawing has proved superior to other systems at the Glenn L. Martin plant. The development versions of the new machine and the new instruments have been used and found invaluable in simplifying and speeding up the preparation of trimetric drawings. At present the equipment is being used on drawings for the JRM-1 Mars airplane